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ENGINE EFFICIENCY AND EFFECTIVE TURNING MOMENTS AND THEIR EXPERIMENTAL DETERMINATION.

Read before the Schiffbautechnische Gesellschaft, Berlin, by HERM. FÖTTINGER.

Translated by ENSIGN CARL A. RICHTER, U. S. N., Member.

The exact determination of the effective performance of marine engines, as soon as it dealt with amounts of several hundred horsepower, was impossible up to a short time ago. The difficulties of a brake, called forth by high powers, the constricted spaces, the inaccessibility of the engines, of the propellers, as also of the complete lack of other accurate and also practicable methods of measurement, were the reasons why we had to be satisfied with the indicated performance of the engines.

Still more meager was our information concerning the size, efficiency and course of shaft performances requiring certain

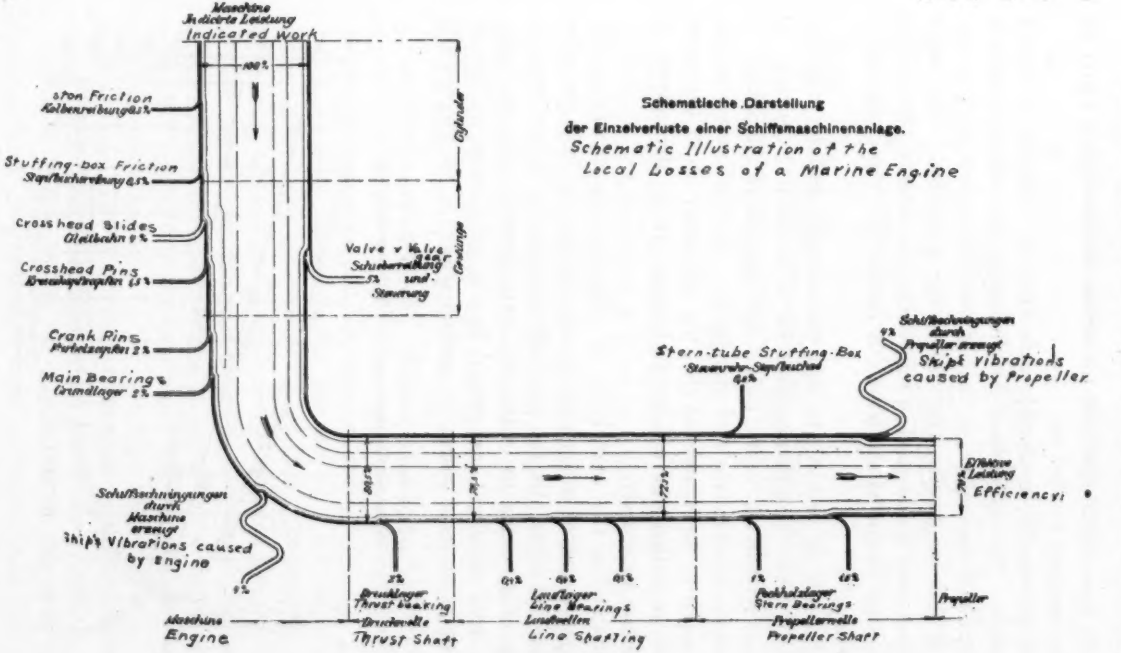
turning moments, and up to now our ideas on these points stand in considerable contrast to the actual performance.

The interesting speed measurements of Dr. Bauer, Stettin, published three years ago, and the more recent torsion measurements of Frahm, Hamburg, and Professor Denton, Hoboken, first brought light in the dark and, unfortunately, often useless, department.

Meanwhile has also the respective theory, partly outstripping experiment, experienced an improvement which nearly all actual performances bear out. I have in mind the work of Professors Lorenz, Gumbel and Frahm.

The methods of measurement and the apparatus which are here discussed owe their origin to my resolution to broaden the practical technical side of this subject, whose imperfection was, in connection with the corresponding propeller computations, disagreeably inconvenient to me. I set before myself the problem of constructing an indicator whose effective performance permitted of determining the actual turning power from a self-produced diagram. The practically obtained solution of the problem is presented in the Torsion Indicator. It is self evident that the road to the solution led through a series of complicated measuring arrangements which in turn unite other advantages.

Before we enter upon the individual methods let us touch lightly upon the difference between the effective and the indicated engine performances or H.P. Indicated H.P. is the work (energy absorbed from the steam by the pistons), which may be easily calculated from the indicator diagram. Like a stream, it flows through the connecting rods, the crank and line shafting to the propeller, its destination. At every bearing along the way a portion of the energy stream trickles away, and of the original stream of energy, the effective I.H.P., only a part is utilized in actually turning the propeller. The numerous losses are indicated in Fig. 1, of which a description would be superfluous. I only wish to emphasize the losses produced by the propellers or the engines in the ship's vibrations, which up to this time have



Schematische Darstellung
der Einzelverluste einer Schiffsmaschinenanlage.
Schematic Illustration of the
Local Losses of a Marine Engine

Fig. 1.

rarely been mentioned, and for whose determination I give an exact method further on.

A universally applicable definition of "efficiency" is not possible. We understand by that term the efficiency of the shaft abaft the engine, or that beyond the thrust shaft, or the stern tube, or, finally, that, up to the propeller. The same arbitrariness holds good in relation to the ratio of effective to indicated performance, which is called the "mechanical efficiency."

It would be valuable in both the scientific and the practical way if the "Schiffbautechnische Gesellschaft" (Society of Naval Architects) would follow the example of the Society of German Electricians and standardize their efficiency terms. It might be recommended that the efficiency up to the stern tube be called the efficiency, because it is the easiest to measure accurately.

Regarding the knowledge of efficiency in specifying the engine power of new ships, in calculating the best propeller sizes, and especially the shaft dimensions, there has been no lack of attempts to utilize the experimental data on hand. For this the ideal way would be the calculation of the individual or local losses, a method which is so successfully

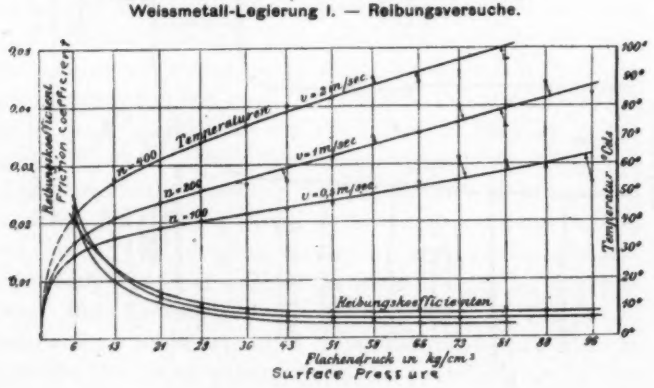


Fig. 2.

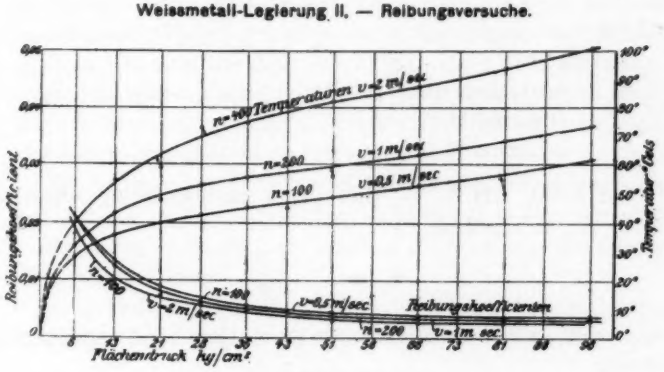


Fig. 3.

applied in the case of modern dynamos and motors that we can calculate beforehand as close as ½ per cent. the probable efficiency of larger types.

In the case of marine engines the usual way is to calculate the individual friction from the total pressure at the respective parts and from the friction coefficient. Until now the friction coefficients for bearings and the like were taken from the special experiments in which the direction and the amount of pressure were kept constant. Figs. 2 and 3 show, for example, the very great decrease in friction coefficients at high pressure with modern white-metal bearings and steel shafts.

Unfortunately, however, the journals of a marine engine work under entirely different conditions. Because of the alternating strokes the journals are lifted from the bearings twice in each revolution, and the oil is drawn in from the oil ducts, and varying conditions of oiling and friction undoubtedly enter largely into the question. The lack of knowledge of what really happens in the bearings, modern valves and stuffing boxes, and of the energy absorbed in the ship's vibrations, makes the preceding method practically worthless.

Of more importance is the calculation of the work done under no load. For stationary engines, this formula is of use:

$$N_i = N_l + \delta . N_e + N_e,$$

where

N_i = I.H.P.;
N_l = work done under no load;
N_e = effective H.P.;
δ = a constant.

For every I.H.P., N_i, there is a corresponding effective H.P., N_e, in the form

$$N_e = \frac{N_i - N_l}{1 + \delta},$$

and that also, with constant revolutions and alternating stroke.

In the case of marine engines, however, each stroke corresponds to a different revolution speed. We have at hand rough assumptions as to the relation between power and revolutions, and have attempted to apply this relation to the marine engine. The formulas obtained in this way are highly complicated and lack, at present, experimental proof in the case of modern marine engines.

An interesting way by which we may obtain an insight into the rate of work with alternating strokes has been devised by French engineers. Working on one shaft are two engines independent of each other, as is the case with some American and French warships, and the S. S. *Kaiser Wilhelm II*, and these are run so as to obtain the same speed with both or with either one. In the first case, where each engine did half the work, the I.H.P. per screw was found to be greater than in the second. In this way the above mentioned formulas were tested.

This method is generally possible. The same number of revolutions can be split in any desired way between the forward and the after engines; in fact, we can go so far as to work one engine backwards—that is, to oppose one engine to the other so that only the difference of their turning moments will be available at the propeller. In principle this experiment is possible, even to the extent of applying the full power

of the one engine against that of the other, with the propellers uncoupled. The excess of power of one engine over the other necessary to turn the shaft would give the I.H.P. necessary to overcome the combined friction of both engines at full power; that is, we would obtain directly the indicated sum total of the losses, and thereby the effective H.P.

The extraordinary danger of this last operation would forbid such a highly interesting experiment being attempted.

The oldest experimental method of obtaining the effective H.P. is the brake.

As is well known, the power developed is directly measured by a scale and the energy is transformed into heat.

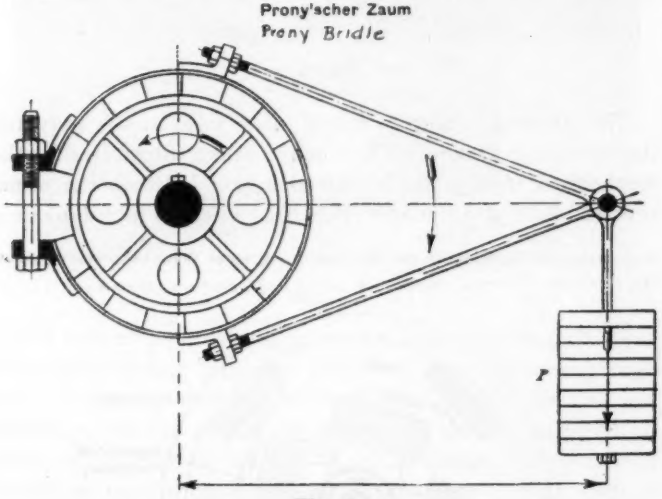


Fig. 4.

The most primitive form of brake, the Prony bridle (Fig. 4), I may assume to be well known, the principal disadvantages of which, the irregular friction and the considerable wear on the bridle, are cleverly avoided in the brake devised by Siemens and Halske (Fig. 5). At the first tendency to burn, the tension is automatically lessened by the working of an inner lever.

Adjustable Brake with Automatic Heat Preventer
Regulirbare Bandbremse mit selbstthätiger Verhütung des Festbrennens.

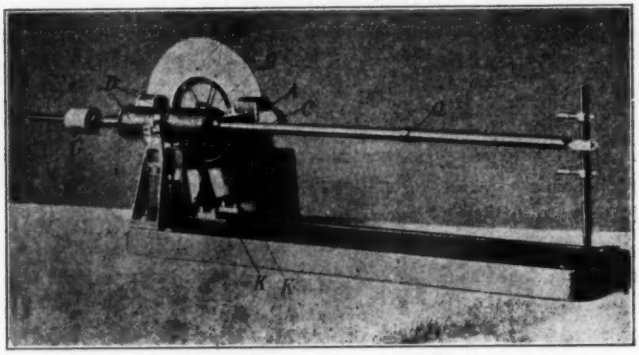


Fig. 5.

The question is prettily solved other ways by the eddy and the hydraulic brakes. The "eddy" brake substitutes for the mechanical friction the electrical action between the armature and field of a dynamo (Fig. 6). There is in a dynamo a

Schematische Darstellung der Bremswirkung einer Wechselstrommaschine.
Diagram of the Working of an Electric Brake.

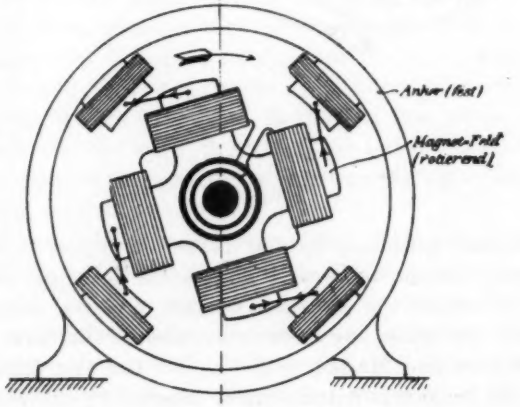


Fig. 6.

strong resistance to turning as soon as the circuit is closed. If the field were free to revolve about the armature it would soon take up the same number of revolutions as the armature; this turning of the field is prevented by a weighted lever, and the turning moment or B.H.P. is thereby measured.

This idea is worked out in the electric brakes of Siemens and Halske (Fig. 7) and of E. H. Rieter (Figs. 8–10).

Eddy Brake (Large Model)
Wirbelstrombremse (grosses Modell).

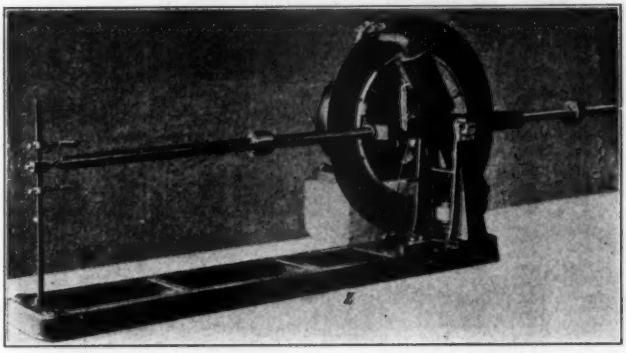


Fig. 7.

The former has a rotating copper disc, the latter an iron drum with cooling ribs, and they both have stationary magnets. A special covering or insulation for the field is not necessary, for the current generated is better dissipated in eddies on the metallic surface of the armature. Any desired number of revolutions is obtained by altering the strength of the field current. The figures show clearly the levers for measuring the torque.

Froude's hydraulic dynamometer absorbs the power by means of violent water eddies. A water wheel rotates in an enclosed water-filled casing. The wheel and the casing have corresponding annular channels of semi-elliptical cross section, the channels being divided into small cups by diagonal partitions. When the shaft rotates the water in the wheel cups

Exact Brake Dynamometer
Präcisions-Bremsdynamometer von C. H. Rieter.

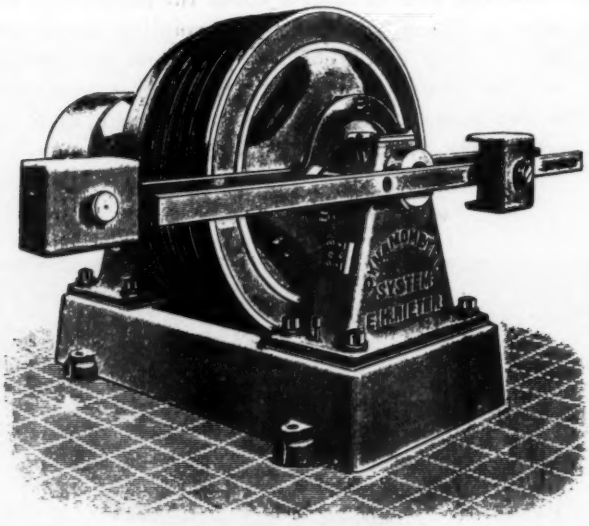


Fig. 8.

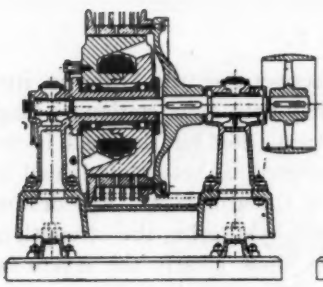


Fig. 9.

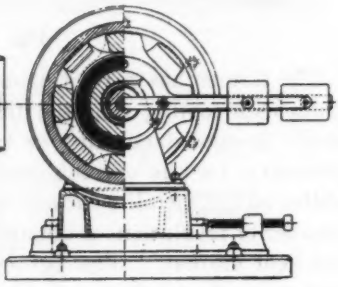


Fig. 10.

is thrown outward by centrifugal force, is sucked on near the shaft and is so forced on with increasing agitation.

The casing is movable about the shaft, and the action of the water makes it tend to rotate with the wheel. This tendency is balanced by a weighted lever, and the torque

thereby measured. The clearance spaces are adjusted by distance pieces on the horizontal screw studs. The firm of Heenan & Froude, Birmingham, advertise a water dynamometer with a 770-mm. water wheel, which will measure 1,100 B.H.P. at 350 revolutions, which is about the power of a small torpedo boat's engines.

A brake necessitates expensive and detailed preparations and measures only the average torque, while the widely vary-

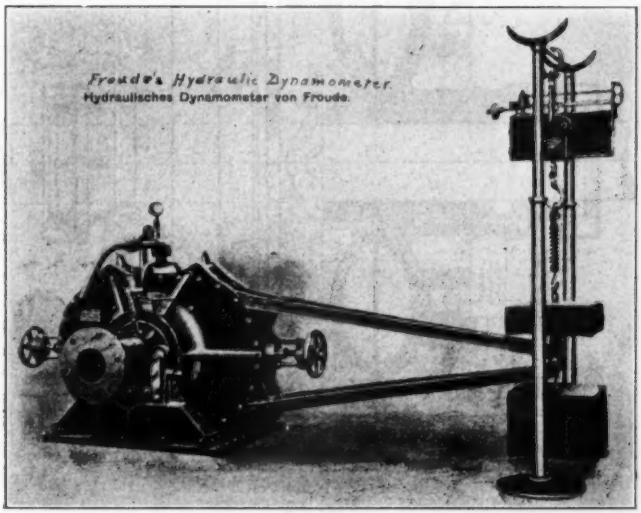


Fig. 11.

ing torques, as found in marine engines, are only qualitatively indicated by a brake in the form of oscillations.

So far, in order to measure and register the variations in torque during a revolution, we had to have recourse to an inserted dynamometer. In this the shaft is broken up, and an arrangement devised which, by means of weights or springs, determines the torque.

Of the many devices of this type there are only two which are applicable to marine engines. White's cogwheel dyna-

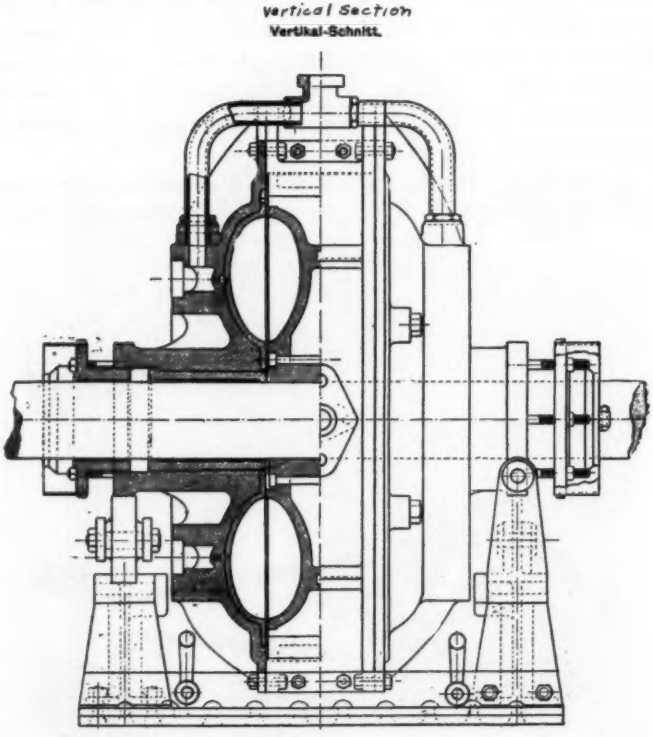

Vertical Section
Vertikal-Schnitt.

Fig. 12.

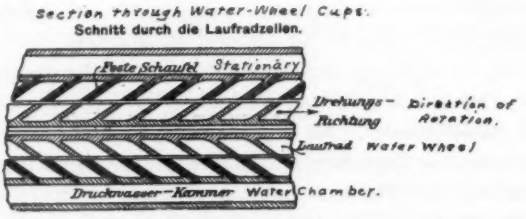

Section through Water-Wheel Cups.
Schnitt durch die Laufradzellen.
Feste Schaufel Stationary
Drehungs-Richtung Direction of Rotation.
Laufrad Water Wheel
Druckwasser-Kammer Water Chamber.

Fig. 13.

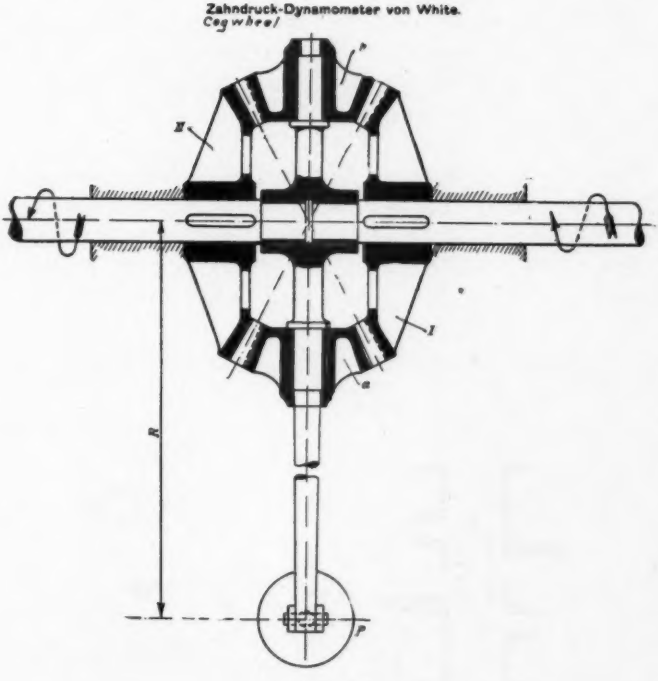


Fig. 14.

mometer is shown in Fig. 14. The entire wheels *a* and *b* are loose on an arm, which is free to revolve on the shaft and transmit the torque from *I* to *II*. It is plain that the arm must be kept from rotating, otherwise it will not transmit any power from *I* to *II*. A turning moment of $P \times E$, equal to that of the shaft, is therefore necessary to hold the arm stationary.

A weighted lever would not be applicable, on account of the noise and jolting of the spur wheels.

By replacing the weights with springs the instrument is made self-adjusting, but with this there are difficulties in dealing with more than 100 H.P., because the direction of revolution of the shaft is reversed.

The torsion dynamometer shown in Fig. 15 acts on an entirely different principle. Power is carried from one shaft to the other by means of spiral or flat springs, whose distortion at any moment indicates the torque, as is done in the case of the indicator.

The principal difficulty of this kind of apparatus lies in observing and registering the relative torque of the two sections of shafting.

Ayrton and Perry's, Fig. 15, shows a small lever K with a shining knob at the end, which turns outward as the torsion increases, and a fixed rule permits us to easily, if not very accurately, measure the diameter of the circle swept by the

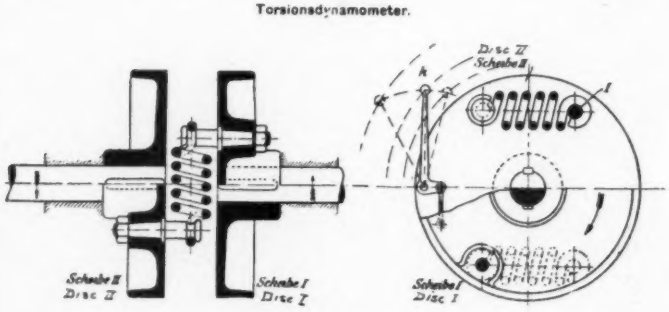


Fig. 15.

knob and thence the torque. With this apparatus the indication of the torque during one revolution is not possible. We will find the same difficulty in the case of the torsion meter, but it has there been overcome in a simple manner.

Although the inserted dynamometers give much better results than brakes, the value of the results is not at all proportionate to the expense and the danger. A broken spring means a broken shaft.

There is known only one case in marine engine practice, that of the *Turbinia*, where an inserted dynamometer is used.

In the determination of the torque during one revolution this kind of dynamometer gives false results. The experi-

mental researches of Frahm, as well as my own (Föttinger) demonstrate irrefutably, and in accordance with theory, that all the torques applied to the shaft are there affected by the torsional vibrations brought on by the crank and the propeller.

The insertion of the elastic dynamometer alters the stiffness and thereby the torsions of the shaft, so that the character of the normal impulse is entirely changed, assuming that the dynamometer is not normally inserted.

Since we are not permitted to change the nature of the shaft, there remains only one applicable method, that of obtaining the torque from the torsion of the shaft, a method that was introduced sixty years ago by the distinguished student of thermo-dynamics, Hirn, and in later years by Denton, Frahm and myself (Föttinger).

For all kinds of forged iron and steel there is a definite relation between the actual torque and the angle of torsion. The following equation holds good:

$$S = \frac{L \times R \times M}{G \times \theta} = \text{const.} \times M,$$

where

S = the measured arc of torsion in Cm,

R = radius of arc S also in Cm,

L = length in Cm of shaft over which the torsion is measured,

G = modulus of elasticity of the shaft in Kg/cm^2,

θ = polar radius of gyration of the crosss ection of the shaft in cm^4, and

M = actual torque in cm/Kg.

Therefore, if we measure the circumferential torsion S, we need only multiply by a constant to find the effective turning moment. We must know, of course, the modulus G, which may be accurately ascertained from experimental shafts or from the actual shaft.

Following is an example of measured torque:

With a shaft 25.2 meters long, 320 mm. diameter, transmitting 2,000 H.P. at 76 revolutions, the mean arc of torsion 315 mm. from the center of the shaft was 17.5 mm.

To obtain the maximum arc of torsion the radius R of measurement, at the length L, must be as large as possible. While it is easy to measure the torsion of a stationary laboratory shaft with aid of a magnifying glass, there are certain difficulties in the case of a rotating shaft.

PRINCIPLE OF ALL TORSION MEASUREMENTS.

The principle of all torsion measurements is shown in Fig. 16. On the two discs a and b, are scribed two radii which are parallel when the shaft is not subject to torque. When torsion exists, these radii form an angle with each other. For example, when the radius b has just reached a vertical posi-

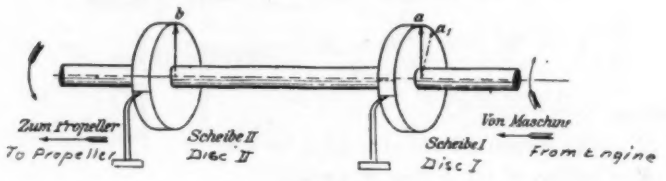


Fig. 16.

tion, the radius a has already passed beyond the vertical to the position A_1; for the rear disc is held back by the resistance of the propeller. That is, while the shaft is doing work the forward disc is always ahead of the after one by the angle of torsion.

We now assume that with a ship's shaft at rest we have scribed a zero mark on the forward and one on the after disc, and that it is possible at any instant, with the same scribing arrangement, to scribe two new marks while the shaft is working.

If the shaft were perfectly rigid and rotated without torsion

the new marks would necessarily be an equal distance from their respective zero marks.

Actually, however, the zero mark on the forward disc precedes that on the after disc by the length of torsion arc, so that the new marks will be found unequally distant from their respective zero marks by the length of torsion arc.

If we subtract, then, from each other the distances of the new marks we will have the arc of torsion at that instant. This differential principle underlies all methods of torsion measurement.

The following developments of this principle differ only in the way by which the marks are obtained.

METHOD OF SYNCHRONOUS TUNING FORKS.

This method was used by Dr. Bauer in his speed measurements. Two paper-covered drums were fixed to the shaft at a known distance apart. Two electro-magnetic tuning forks vibrated by the same current scribed wave lines on the paper. The speed of the shaft was determined from the length of the waves. By means of a simple arrangement this method may be used for torsion measurements. The two tuning forks may be made synchronous, keep step with each other, by

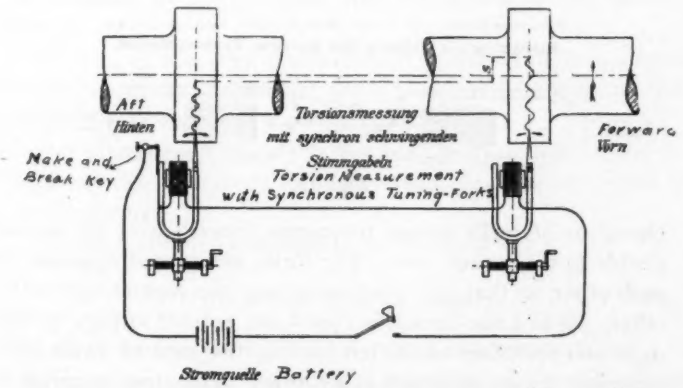


Fig. 17.

using one make-and-break for both. This arrangement is shown in Fig. 17. If the forks have the same rates of vibration they will, when actuated by the same current, strike out their arcs simultaneously. If the forks strike these arcs on the paper with a pencil, their intersection with the wave lines will give us a means of measuring the torsion.

Before the experiment the zero marks must be put on the drums for some definite crank position, such as "beginning of stroke."

Since the shaft always lies with an appreciable amount of torsion in the line bearings, stern bearings and stern-tube stuffing box, the zero mark is obtained by turning once forward and once backward, making a mark for each, and taking the point midway between these two for the real zero.

The amplitude of the vibrations of these forks was very small and caused the waves to be very long and flat, so I armed the prongs with extensions which actuated a small lever. The amplitude was thus multiplied fourfold.

There will not be true synchronism unless the frequencies of all the prongs are exactly alike. The frequencies may be adjusted by careful filing or by attaching very small weights.

I cannot enter into results of these interesting oscillation observations, but indicate the way (Fig. 18) to make syn-

Arrangement to test for exact Synchronism
Anordnung zur Prüfung des genauen Synchronismus.

Fig. 18.

chronism of forks whose frequency exceeds 100 per second visible to the naked eye. The forks are placed opposite to each other, so that in a position of rest one pointer covers the other. When the circuit is closed one pointer swings to the right and the other to the left, and in the case of exact synchronism always pass each other at *m*. The crossing point *m* is easily observed in the veil formed, and with the least

change of phase it shifts to one side or the other. This method has even been used to demonstrate the phase variations of alternating currents.

Another method of testing for synchronism consists in making each pointer mark on a rotating drum, and regulating the distances of the marks from the starting point.

I had first resolved upon this tuning-fork method in connection with the torsionmeter diagram, but for lack of time I adopted, at the last moment, the use of electric flashes, which required less preparation and which I shall describe later.

METHOD WITH TWO SYNCHRONOUS MOTORS.

The difficult synchronizing of the two recording pencils is avoided by using small rotating synchronous motors. In connection with their principle it will be remembered that with electrically-controlled apparatus an exact motion like that of the hands of a clock is obtained by the use of certain electric arrangements like that of a revolving field.

If, with a revolving field, we did not restrict the pointer to one turn, but permitted it to make several, there would have to be an equal number of marks on the drum. If we put two motors in series they would revolve synchronously, coupled, so to speak, by electric force, like the hands of two electric clocks.

In Fig. 19 it is seen that with every rotation there is a mark made on the lamp-blacked paper; the circumferential line may thereby be cut at a good angle.

An alternating current is not always available, so the direct current of a ship's lighting system may be converted in a transformer.

A much simpler principle is embodied in the

METHOD OF ELECTRIC FLASHES,

which I have applied with success to the torsion meter. If we insert two spark gaps in the secondary of a spark coil instead of the customary one, the spark will jump across both

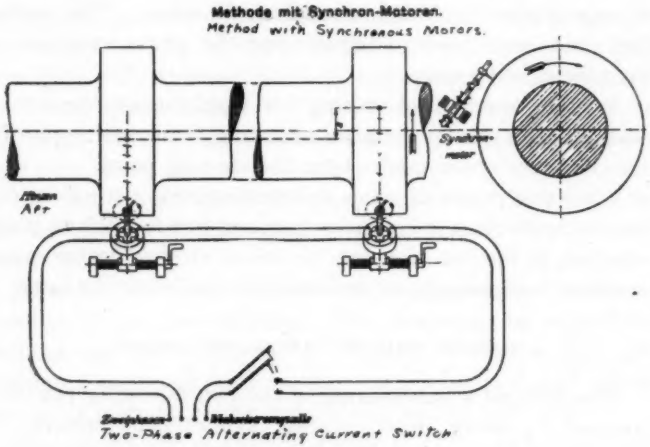

Methode mit Synchron-Motoren.
Method with Synchronous Motors.
Two-Phase Alternating Current Switch

Fig. 19.

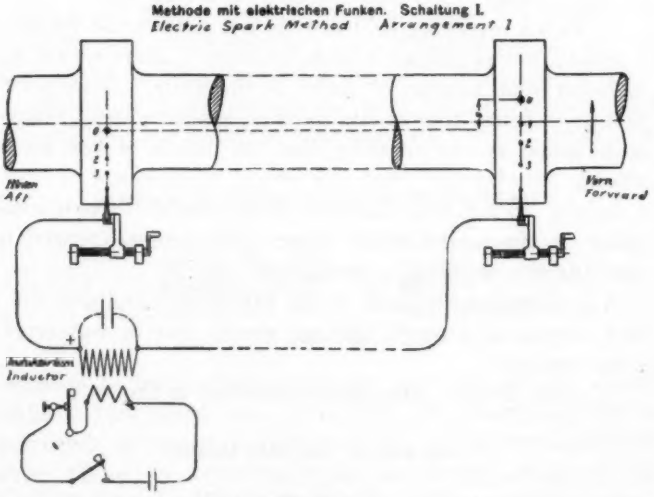

Methode mit elektrischen Funken. Schaltung I.
Electric Spark Method Arrangement I
Forward
Inductor

Fig. 20.

gaps at exactly the same instant and perforate a sheet of paper inserted in each opening.

This simple idea is shown in Fig. 20. The two poles of the secondary of an induction coil consist of two metallic points which are distant about 1 mm. from their respective drums or collars. The shaft then forms part of the circuit. As soon as the interrupter begins to work sparks jump across the two gaps to the rotating paper-covered collars and make distinct holes in the paper.

A variation of this method is shown in Fig. 21. There are here two induction coils, whose primaries are in series with each other and are actuated by the same interrupter. The

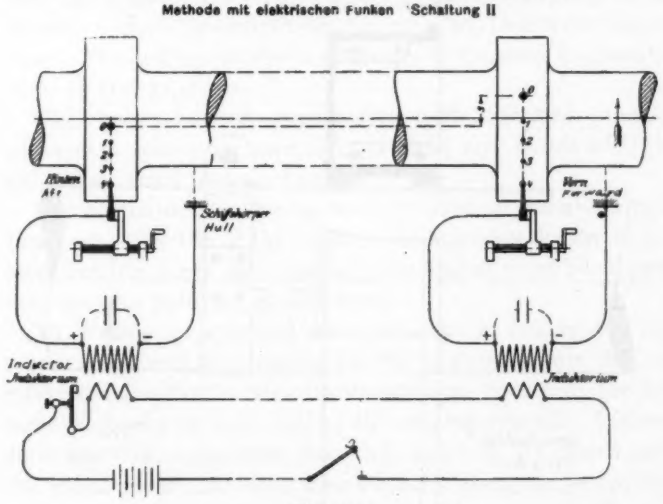


Fig. 21.

positive poles of the secondary are connected with the metal points, and the negative poles to the shaft (and thereby the ship's structure). As before, when the interrupter operates, sparks jump across their respective gaps and perforate the paper on the collars.

Both arrangements have been successful in practice. The second has the advantage of having no wiring between the two

secondaries and being more easily isolated in the shaft alley than the first. On the other hand, two inductors and a strong battery are necessary. To obtain very distinct holes Leyden jars are inserted in the secondary circuit.

To locate the holes easily it is advantageous to use paper blackened by a turpentine flame, for the spark blows the black deposit away for a radius of about 1 mm., so that the points appear to be white.

To keep the metal points the same distance away from the paper a polished hard-rubber ring was mounted on the collar, and this made a white line along the row of holes.

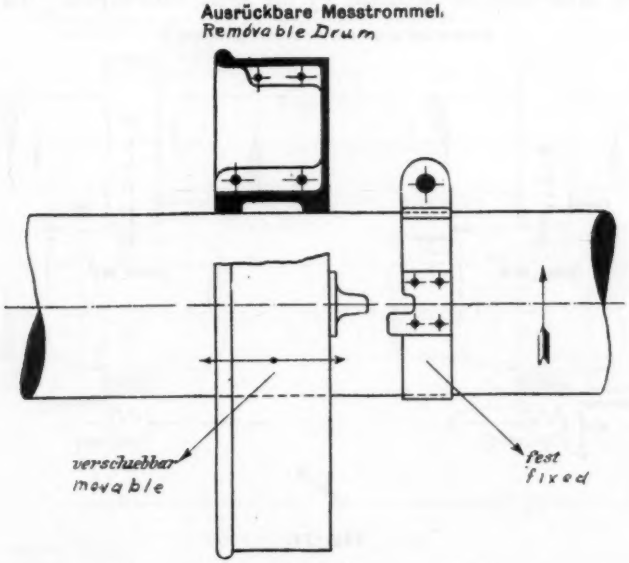


Fig. 22.

In order to avoid the trouble and loss of time necessary in locating the zero marks for each diagram, and to avoid stopping the engine, special removable measuring drums, 764 mm. diameter = 2,400 mm. circumference, were used, Fig. 22. The adjacent collar attached to the shaft has a wedge-shaped opening in which *a*, the wedge on the movable drum may be

inserted. By this arrangement an exact coupling between drum and shaft is obtained, and there is the advantage of being able to attach any desired number of papers to the drum while the shaft is revolving.

It is assumed that the zero mark has been permanently and accurately marked on the drum as well as on the diagrams.

There is the further advantage of being able to select a drum large enough in diameter to give a much larger arc of torsion than could be obtained on a shaft collar.

FRAHM'S METHOD.

This utilizes in an entirely different way the electric current for a simultaneous recording of the marks, and is described in the "Zeitschrift der Vereinigten Deutscher Ingeneure" (Proceedings of the Association of German Engineers), Nos. 22 and 24, 1902.

The chemical action of an electric current is used to make permanent marks on specially prepared zinc bands with the aid of two fixed platinum points.

For further details the reader is referred to the above publications. The use of the shaft collar increases the arc of torsion, but, for every new diagram, the engine must be stopped and the zero point be located anew.

Mr. Frahm experienced some difficulty in finding the corresponding rows and marks, for the contact points did not strike the shaft collar simultaneously, and, therefore, the first rows of marks on both collars did not correspond. I, therefore, made the suggestion that the circuit be not closed until the metal points had been moved axially along for one or two revolutions. The first marks on the two drums will then obviously correspond. This objection against the reliability of the result is thus easily removed.

PROFESSOR DENTON'S METHOD.

In regard to this I can only state that the efficiency of turbines is obtained from the torsion of the shaft by means of electricity. Further details I owe to private information.

Results are obtained from these diagrams by measuring the distances of the marks from their respective zero points and subtracting them from each other. These differences—that is, the torsion arcs—are transferred to another diagram.

If the marks are made at equal time intervals we can calculate the speed of the shaft at any instant from the distance between successive marks. For torsion measurements the length of time interval is immaterial.

The principal objection to the described methods consists in their intricacy. The operation of two measuring instruments with two diagrams, the easy failure of either, the necessity of having three observers, and, finally, the tedious calculations, are a great strain on the patience of the experimenters.

METHOD WITH ONE DRUM.

This is a distinct advance in torsion measurements. The differencing of the torsion marks is taken care of by the apparatus itself. The after drum carries a metal band having

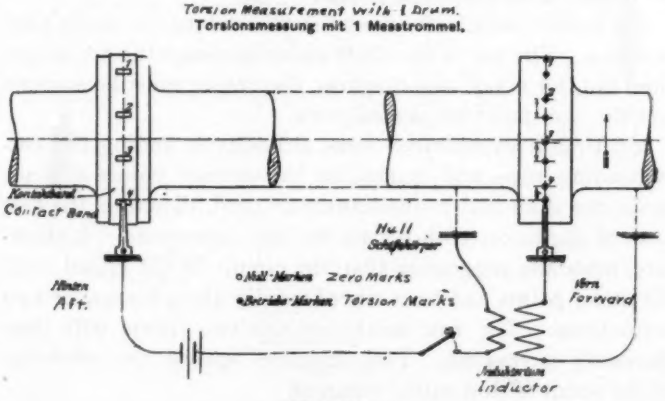


Fig. 23.

a series of insulated spots, 1, 2, 3, 4 (Fig. 23). As the shaft revolves these spots pass under a brush and thereby break the circuit of the primary of an induction coil located near the forward measuring drum.

At the instant of each break there is produced in the secondary circuit a spark which perforates the paper on the forward drum. Then, by slow rocking back and forth of the shaft, the zero marks corresponding to the breaks are located.

If the shaft were to turn without torsion the spark points would all fall exactly on the mean zero points. Actually, however, the working marks fall some distance from their zero marks, and the torsion for each position at no load can then be measured.

It is to be noticed in practice that in going ahead one edge of the break is the effective one, and in going astern the

Anordnung einer besonderen Rückwärtsbürste.
Arrangement with Special Backing Brush

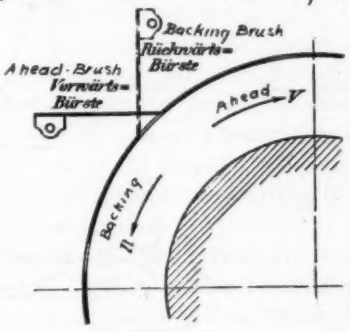


Fig. 24.

opposite edge is effective, and, therefore, the ahead and astern zero marks do not coincide.

This difficulty is avoided by a two-brush arrangement, shown in Fig. 24, by means of which the ahead break is made at the same instant as the going-astern break, and their zero marks would therefore coincide. The backing or astern brush is taken up after its zero point is made.

In the experiments there is nothing to do beyond placing the contact brush and the metal pole, whereby the circuits are completed, and the metal pole is then gradually moved axially along its drum. As before, the removable drum may

be used here to obtain several diagrams without stopping the engine.

An interesting variation (Fig. 25) of this method consists in making the torsions, and thereby the torques, visible by means of a luminous curve caused by the sparks. The spark gaps in this case are on the drum in a spiral curve around the drum.

The tangential distance (the ordinate) of the sparks from a given zero position is proportional to the torsion arc; the axial movement (the abscissa) of the spark is proportional to

Direct Representation of the Torsion Curve by a Chain of Sparks.
Direkte Darstellung der Verdrehungskurve durch eine leuchtende Funkenkette.

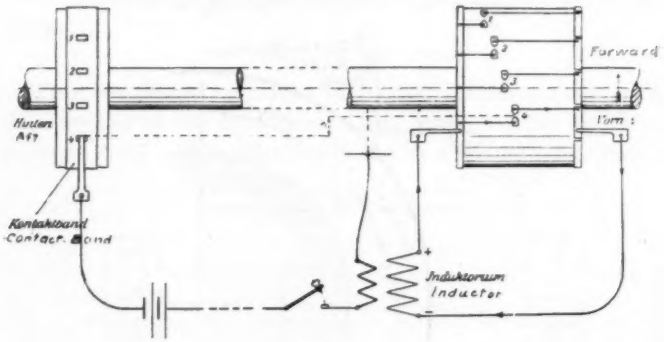


Fig. 25.

the angular twist of the forward drum. With the aid of a pane of glass the curve for each revolution can be traced on tracing paper.

I would have applied this method to a model if I had had it in time.

The use of the diagram by this last method and modification is very simple, and the torsion arc at any instant can be taken off directly.

I had intended to determine the torsion in this way on board a cruiser, but had, unfortunately, to satisfy myself with the calculation that the available shaft length at 10,000 H.P.

would give only 3 mm. torsion at the circumference of the collar. To calculate the torque from so small a measure is practically impossible; one needs at least a torsion of 15 mm., that is, a shaft length of 20 to 30 meters. This is a serious difficulty, one that shuts out from warship use all the foregoing methods.

There are also other difficulties. A shaft of that length consists of 4 or 5 sections and has 8 or 10 collars, concerning whose torsional behavior practically nothing is known.

Since the distribution of load along the shaft is not uniform, in the case of a long shaft we are reduced to inaccurate calculations, whose errors extend to all further results.

Moreover, such a shaft would necessitate 8 to 10 bearings, each one of which absorbs some energy.

In order to have unobjectionable results, the length of shafting used must be free from flanges and bearings. We are therefore restricted to the length of shaft between any two bearings, often only 2 or 3 meters in length.

In the case of a merchant steamer, a 2-meter shaft length, at a radius of 400 mm. gave a torsion of only 1.8 mm., with 2,000 H.P. and 76 revolutions.

Since in all similar cases the torsion arc is only 1 to 3 mm. (in the case of small warships it may be 6 to 8 mm.), it follows that improvement must follow not on the observer's end, but on the machine's end of the measuring. Otherwise, the errors of observation would often exceed the actual measurement.

The principle of torsion measurement with short shaft lengths (1 to 2 meters) is shown in Fig. 26. There is clamped to the shaft a sleeve which is free at its other end and carries a disc at the free end. Opposite this disc is another made directly fast to the shaft. If torque is applied to the shaft the two discs have a motion relative to each other in proportion to the torsion angle. The torque might then be measured from the divergence of two corresponding points on the discs. Every change in torque would cause a corresponding movement of the corresponding points.

For practical measurements arcs of 1 to 3 mm., and even of

Prinzip der Torsionsmessung auf kurze Messlänge.
Principle of Torsion Measurement with Short Shaft Lengths

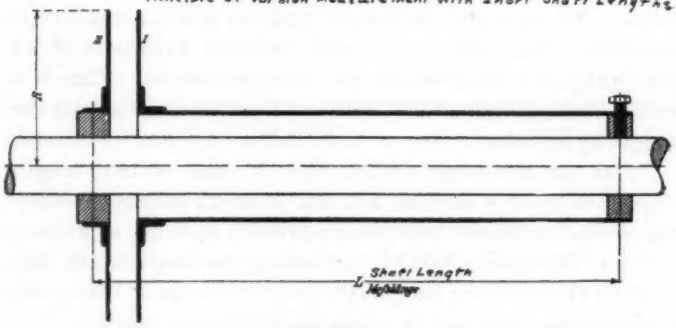


Fig. 26.

6 to 8 mm., are much too small; we may, however, let the measuring gear magnify or multiply this.

The principle of multiplying or magnifying is applied to all the following methods.

There is another point to be observed. The shaft and discs rotate while working while the observer is stationary; we must make the relative movement of the discs quantitatively visible.

In an extremely simple way both points are solved in the

Enlargement and Transfer of the Torsion Arc by Diagonal Slots.
Vergrösserung und Uebertragung des Verdrehungsbogens durch schräge Schlitze.

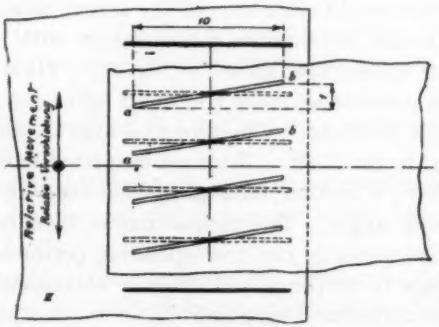


Fig. 27.

enlargement and transfer of the torsion arc by means of diagonal slits.

A white-painted sheet of tin, *I*, has at equal intervals slightly diagonal, parallel slits, *ab*. The tin sheet *II* has at the same intervals horizontal black lines (Fig. 27). If *I* is superposed on *II* a very small portion of each black line is visible through the slits. Every relative movement of the plates in the direction of the arrows causes a distinct movement of the visible black portion between *a* and *b*. If the angularity of the slits is 1 to 10 the movement of the black spots is ten times that of the plates. The action is the same if the tin plates are rolled on two concentric cylinders whose axes are parallel to the black lines.

If the cylinders rotate, the black parts show as a dark ring on the white cylinder covering, and if we apply this arrangement to Fig. 26, every relative movement of the discs will cause an axial displacement of the black ring. The distance of this black ring from the zero point measures the torsion arc and the play back and forth of the ring represents the variation of the effective torque.

This arrangement is not to be surpassed for cheapness and simplicity, and permits the maximum and minimum torque to be read on a scale without any further complications. To obtain more points for the construction of a curve, a screen rotating with the shaft is used. This covers all but one or two of the slits, and may be moved through very small distances.

Enlargement and transfer of the torsion arc by means of a ray of light affords some finer measurements.

The discs *I* and *II* are perforated near their circumference by holes that are opposite each other when there is no load on the shaft (Fig. 28). At the height of the shaft axis a fixed arc light throws only a horizontal ray through two of the holes, thereby giving a zero point on a vertical scale. When the shaft is working the relative movement of the discs causes the line joining any two corresponding holes to be diagonal, and the light ray follows this line. Therefore the

spot of light on the scale is moved from the zero point through the distance S, which is proportional to the torsion s. As the shaft rotates, each pair of holes causes the spot of

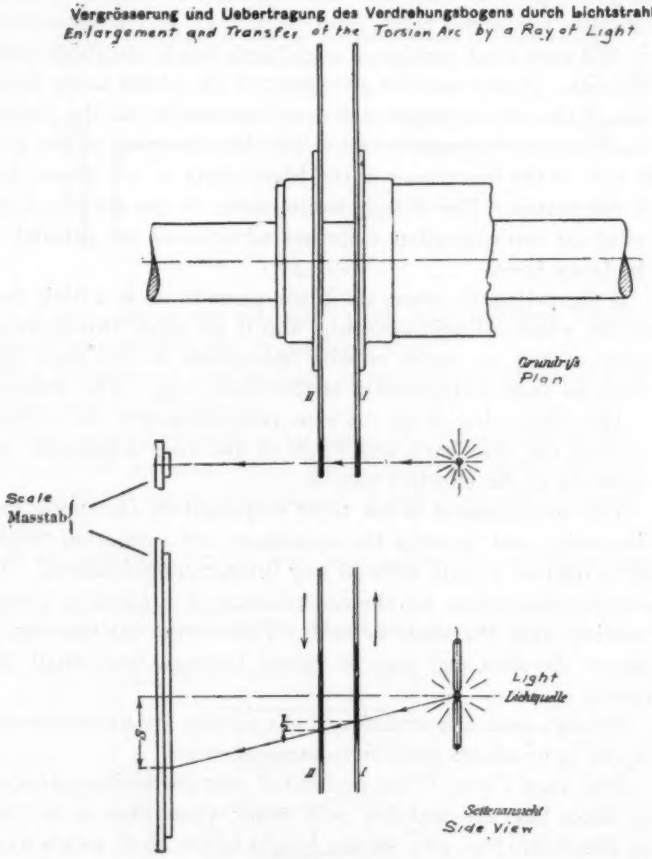


Fig. 28.

light to dance up and down the scale, picturing the variations in the torque.

In practice the light must be concentrated by means of

lenses and diaphragms. For the reading of single-torsion values a rotating screen may be used to advantage.

This system is better adapted to use in physical laboratories than in shaft alleys. Technical demands go further. From the technical standpoint it is not satisfactory to tediously get results from measurements obtained with difficulty. On trial trips, for example, the engineer wants to see immediate results in order to handle the engines to best advantage.

What is wanted is an arrangement that will give a diagram indicating the effective torque without the necessity for any mental effort.

THE TORSION METER.

This furnishes a solution of this comprehensive problem and is simple in principle. The small relative movements of the discs is multiplied by one or more levers in exactly the same way as is done with all steam indicators and manometers.

The principle difficulty lay in making the diagram obtained from the rotating discs easily accessible. To stop the engine is practically out of the question. A surprisingly simple solution is shown in Fig. 29. The levers *a*, *b*, *c*, *d*, *e*, *f*, *g*, are so

Pencil-Lever and Drum Arrangement of the Torsion Indicator
Schreibhebel- und Trommelanordnung des Torsionsindikators.

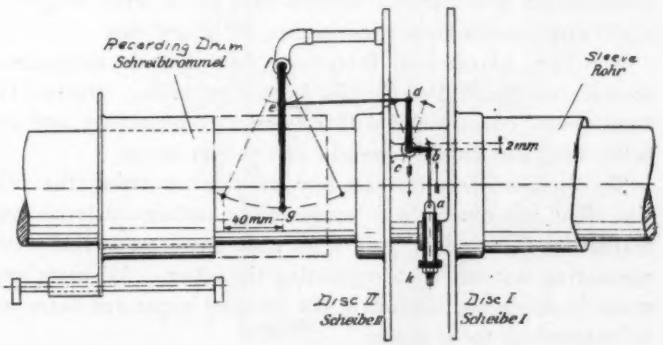


Fig. 29.

arranged that the pencil points to the center of the shaft and moves in the direction of the shaft's axis. Its displacement in this direction from the zero position therefore measures the instantaneous torsion of the shaft and thereby the torque.

A recording drum, concentric with the shaft, is arranged on a sliding drum so that it may be slid under the pencil when desired. In practice, the sleeve, discs and lever system all rotate; the pencil g, therefore, travels around the stationary recording drum and leaves on it the torsion curve.

When this drum is slid to one side, it is entirely free from the rotating part of the apparatus, and may be safely mounted with paper, or the finished diagram removed.

Of course, every diagram must carry a zero line similar to the atmospheric line of the indicator diagram. This is obtained from a second pencil rotating with the shaft, and permanently attached to it when the apparatus is first set up.

The ordinates of the diagram, that is, the axial movement of the pencil, are proportional to the torque at any instant; the abscissae, that is, the travel of the pencil around the drum, are proportional to the distance the ship has traveled; therefore, the area of the diagram, as is the case with the usual indicator diagram, represents the work done by the shaft. It follows that by means of a planimeter the mean torque and thereby the E.H.P. may be obtained. Figs. 30–34 show the constructive development of this idea for a shaft length of 2,200 mm., and a measuring radius, R, of 400 mm.

A cursory glance will betray the fact that the elegance of modern mechanical skill has been cast aside. Rather the details were constructed with reference to reliability and stability with the smallest weight and proper shape.

To discover and eliminate any one-sided working the relative disc movement was measured simultaneously at two diametrically opposite points, so that there were really two measuring systems, each regulating the other. All parts were made in duplicate, so that in the finished apparatus there are no corrections to be made.

The sleeve is centered on the shaft by means of four large

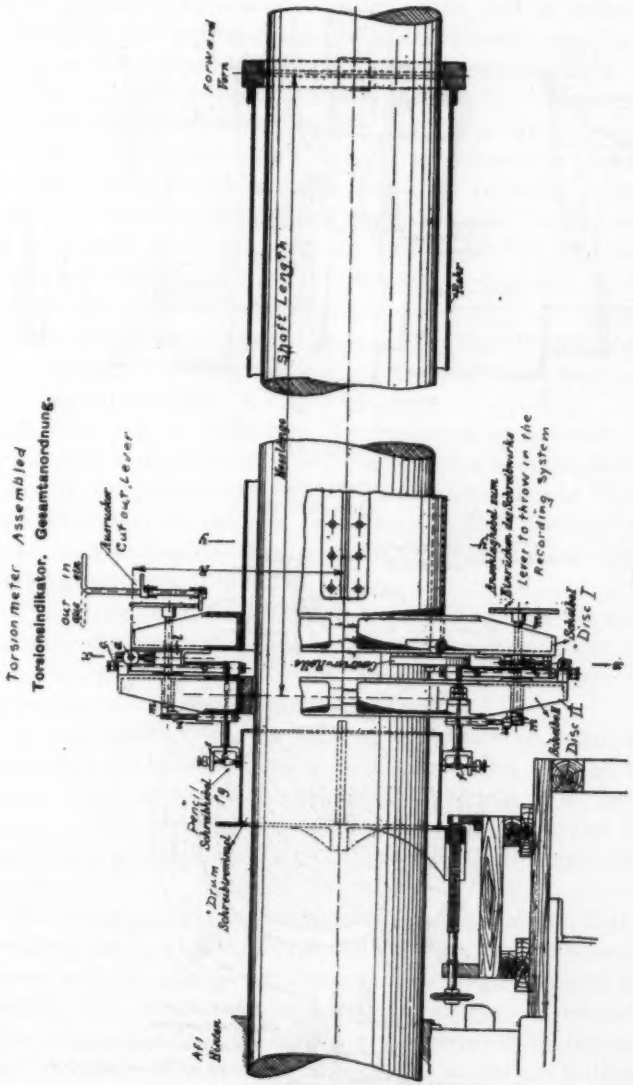

Torsion meter Assembled
Torsionsindikator. Gesamtanordnung.
Forward
Vorn
Shaft Length
Kurbelwelle
Ausrücker
Cut out Lever
Aus
Ein
Hebel zum Einrücken des Schreibwerks
Lever to throw in the Recording System
Scheibe
Disc I
Scheibe
Disc II
Profil Scheibe
Profil Disc
Trommel Schreibwerk
Drum
Aft
Hinten

Fig. 30.

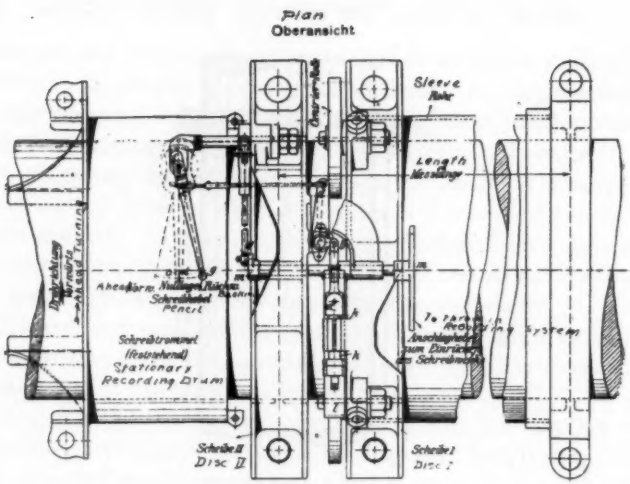

Plan
Oberansicht
Sleeve Roller
Length Measuring
Schreibstift Pencil
Schreibtrommel (feststehend) Stationary Recording Drum
Schreibscheibe II Disc II
Schreibscheibe I Disc I

Fig. 31.

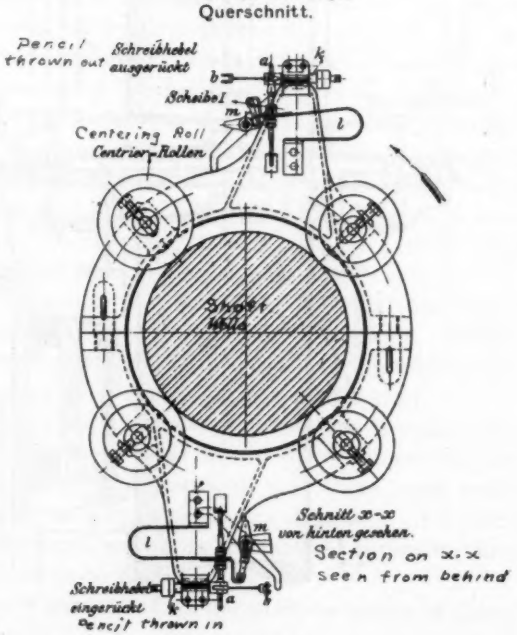

Cross Section
Querschnitt.
Pencil thrown out
Schreibhebel ausgerückt
Scheibel
Centering Roll
Centrier-Rollen
Shaft Hole
Schnitt x-x von hinten gesehen.
Section on x-x seen from behind
Schreibhebel eingerückt
Pencil thrown in

Fig. 32.

balls, adjustable by means of radial screws, and by means of these balls the friction is reduced to a minimum. In the arrangement for a 50-mm. shaft diameter roll sectors were used instead, because of irregularities on the shaft.

Following the principle of Fig. 29 the levers *a*, *b*, *c*, *d*, *e*, *f*, *g* (Fig. 31) multiply the relative disc movements 22 times, so that a mean torsion of 2 mm. would be represented by 44 mm. on the diagram. The greatest enemy to an exact measurement is in this case the lost motion in the joints. This was removed as much as possible by means of conical adjustable pins in all stationary parts, and carefully adjusted cylindrical pins for all moving parts. The residual lost motion was rendered harmless by means of a spring that kept each pin pressed in place.

In order not to complicate the apparatus at the start the slide guides of the steam indicator were not adopted, and no error in the measured efficiency resulted, as may be shown geometrically; for the rest, the always slightly distorted diagram may be transferred from polar to perpendicular coordinates, as shown in Fig. 43.

The pencil must be rather firmly pressed against the drum because of the centrifugal force. When the drum is slid into place the pencil would therefore break off if there were not some way of lifting up the pencil.

To accomplish this with the rotating apparatus there is a small spindle *M*, with a crank on one end and a lever on the other, which works on the principle of the star of a ratchet brace. This permits the pillars with the fixed and the oscillating pencils to be turned through a small angle (Figs. 33, 34).

This arrangement also accomplishes another object, that of arresting the lever motion. It did not seem good to have the joints working during the whole run, although one does not consider that point in the case of steam gauges on receivers. The stopping is done by breaking the connections between the two discs. The conical pin *a* (Figs. 32 and 34) is drawn radially inward by a small crank and the spring *l*.

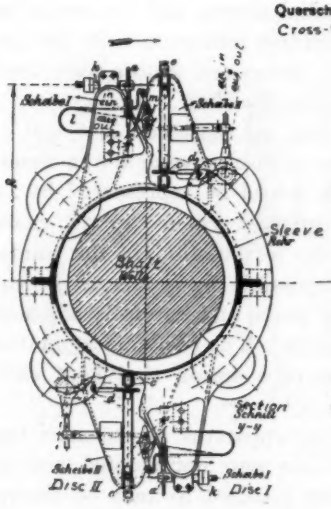


Fig. 33.

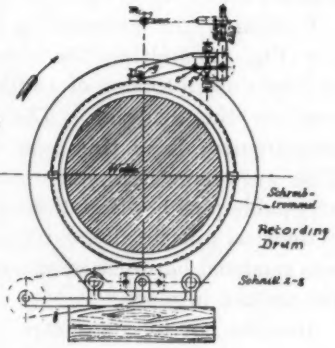


Fig. 34.

The motion of this withdrawing and stopping arrangement is effected by the striking of the right-hand lever (Fig. 30) on either the inner or the outer lugs of an adjustable arm attached to the ship's framing.

One of the chief difficulties in practice is the speed of the recording. With a drum diameter of 380 mm. and 80 r.p.m., the peripheral speed is 1.6 meters per second. With suitable recording arrangements it is possible to obtain good results with even double that speed, so that there is no objection to the use of this torsionmeter either in the largest steamships or in torpedo boats.

In this case a new recording-drum arrangement is used. The principle is shown in Fig. 35. The relative speed of the pencil and the drum is reduced by having the drum rotate with the pencil, but at a slower speed. This is accomplished by the spur gearing q, r, s, which drives the drum, but also permits its being stopped while the shaft is rotating. It is not worth while to go further into the constructive de-

Rotating Drum with Decreased Recording Speed
Rotierende Schreibtrommel mit verminderter Schreibgeschwindigkeit.

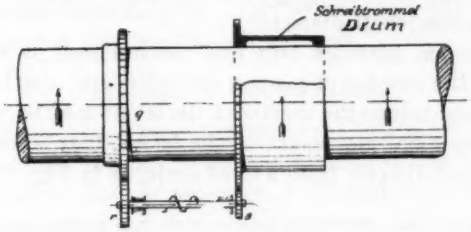


Fig. 35.

velopment of this idea, which has been worked out in an entirely different way by means of small recording drums.

It is hardly necessary to state that the principal idea of the apparatus permitted an endless variety of details, most of which were rejected. Among these was a hydraulic increase in the torsion, and an electrically controlled movement of the pointer at any desired part of the ship.

Possible errors resulting from torsional variations of the sleeve, due to varying torque on the shaft, were avoided, as in Fig. 36. The torsional vibrations of such short sleeves is enormously high.

Arrangement for prevention of Sleeve-vibrations
Anordnung zur Verhütung von Torsionsschwingungen des Rohres.

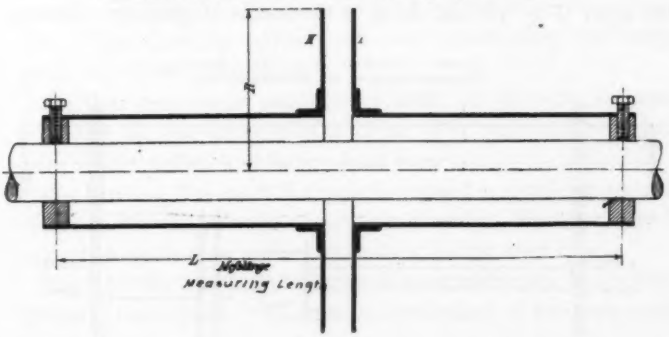


Fig. 36.

If there is little shaft room the recording drum may be placed on the sleeve (Fig. 39), as was done with the model for 50 to 140-mm. shaft diameter.

With steam turbines only one measurement is necessary, owing to the absence of pistons and cylinders; and because of the constant torque the inertia of the levers may be neglected, so that the axial movement of the levers, that is, the torque, may be read directly from a fixed scale, as in Fig. 37.

Torsion Indicator for Turbine Ships with Direct Reading of Torque
Torsionsindikator für Turbinenschiffe mit direkter Ablesung des Drehmoments.

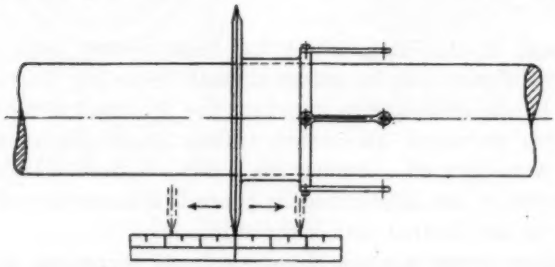


Fig. 37.

The torsionmeter may be attached to smaller shafts by means of an interchangeable case or by means of four set screws.

The apparatus may be made smaller if we separate it from the shaft (Fig. 38) and drive it by means of gearing. In this

Torsion-Indicator with Spur Gearing
Torsionsindikator mit Zahnradantrieb.

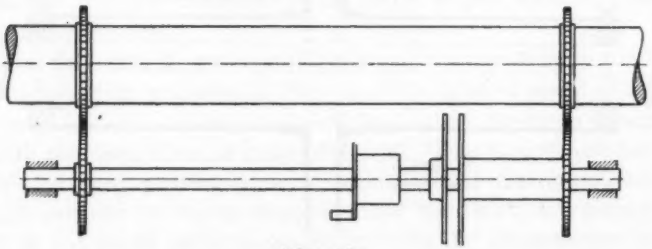


Fig. 38.

case there is a loss due to lost motion and the increase in parts.

The sensitiveness may be varied by changing the radius of measurement or altering the lever system.

HANDLING OF THE INDICATOR.

Before discussing the experiments the method of handling the apparatus is shown. First the position of the pencil for the torque-free shaft is ascertained through slow back and forward rocking of the shaft. Zero marks, some millimeters apart, are thus obtained. Assuming the propeller to be accurately balanced, the true zero line is the mean of these, and there the stationary pencil is fixed.

Then is obtained the position of the pencil on the drum corresponding to the beginning of the piston stroke, by rocking the shaft as before, and taking the mean.

The actual taking of the diagram consists, besides sliding the drum back and forth, in nothing more than turning the cut-out lever (Fig. 30) to the "in" and "out" positions. The entire manipulation was done by a locksmith, given the directions but once.

THE EXPERIMENTS.

The first experiments were made with a rather small model on the power shaft of a punching machine.

Naturally the fairly uniform torque was hardly interesting, because comparisons cannot be made with only one calculation. The starting period only was interesting, and began with very jerky vibrations.

Further tests were made on a shaft of 50 mm. diameter mounted on the centers of a lathe (Fig. 39), and weighted with a belt pulley and heavy lead weights.

On turning the shaft the weight caused a sinuform turning moment, for the pointer moved sinuformly. The curve to be expected was a sine curve symmetric to the zero line.

Fig. 40 shows copies of the curves obtained under four different conditions. They show the course of the torque and the load.

Lathe Test Arrangement.
Versuche auf der Drehbank. Messordnung.

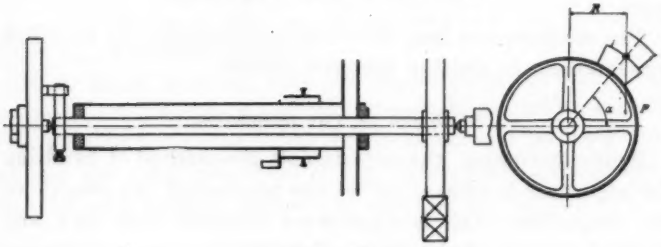


Fig. 39.

The position of the zero line is not exactly symmetrical to the sine curve. This is due to a constant torque necessary to overcome the friction of the after lathe center. By using a planimeter on the diagram the corresponding work could easily be found. By tightening up the lathe center the load could be increased at will.

Even with the low speed of 27 revolutions per minute there are distinct variations from the sine curve near the zero

Lathe Test of a Shaft 50 mm Diameter.
Versuche auf der Drehbank an einer Welle von 50 mm Durchmesser.

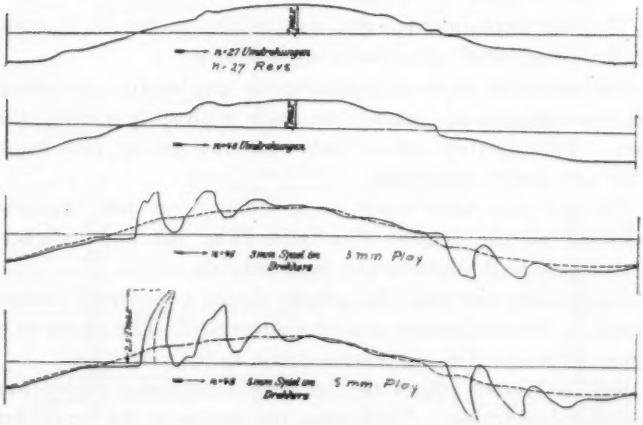


Fig. 40.

line, that is, at the shifting of the load. The cause could be heard in the loud rattling of the spur teeth. The play of the teeth naturally causes vibrations which appear as torsional vibrations of the shaft. This increases with the revolutions, as the second diagram shows.

For special reasons the tests were limited to the cases where there was 3 and 5 mm. play at the dog. The knocking that resulted called forth torsional vibrations which demanded double or three times the maximum static torque. Unfortunately, therefore, the first amplitude after each knock could not be accurately known, since the torsion indicator was not built for such large variations.

The resultant curve is not exactly sinuform because of great torsional vibrations, the more so because of the curved coordinates and of the fact that the abscissae represent angles of advance and not intervals of time. In addition, there are waves on the curve of a still higher period which may be attributed to local vibrations.

From the results far-reaching inferences may be made, for example, in regard to the extent of bending vibrations caused by jolting, and their demands.

If a play of 3 to 5 mm. in an elastic shaft of 50 mm. diameter and 3 m. length requires a power 2 to 3 times the normal driving power, what may we not expect of tail shafts which often have as much as 10 mm. play in the stern bearings and are jolted around by a poorly-balanced propeller?

If we consider that the lateral or bending vibrations are much harder to take up than the twisting vibrations just described, it is easy to see that in the long run the shaft cannot withstand the enormous overstrains.

After the information obtained from the first tests a large indicator was made for a shaft 320 mm. diameter. It was mounted on the three-crank shaft of a merchant steamer of 2,200 I.H.P. and about 75 revolutions.

In spite of the great difference in the powers measured (that of 50 mm. and a 320 mm. shaft), and the fact that the indicator was assembled at the last minute, all parts of it func-

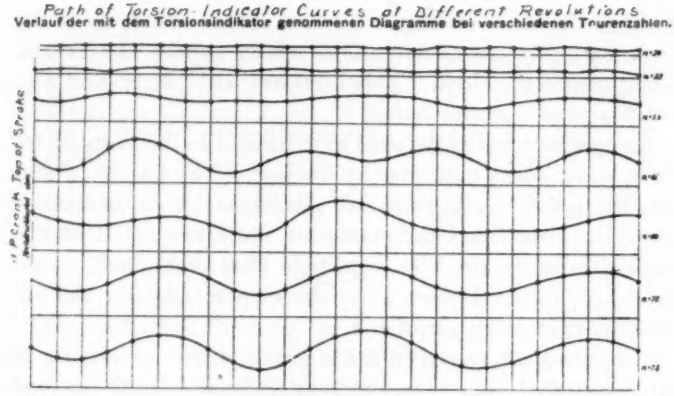

Path of Torsion Indicator Curves at Different Revolutions
Verlauf der mit dem Torsionsindikator genommenen Diagramme bei verschiedenen Tourenzahlen.

Fig. 41.

tioned satisfactorily from the start. Minor accidents at the beginning, such as the flying off of a pencil point or the jarring of the recording lever, do not at all subtract from its complete success.

For example, a series of the original diagrams are shown in Figs. 41–42, taken at different revolutions and referred to curved ordinates. The graphic method of transferring them to rectangular ordinates is shown in Fig. 43. By using the

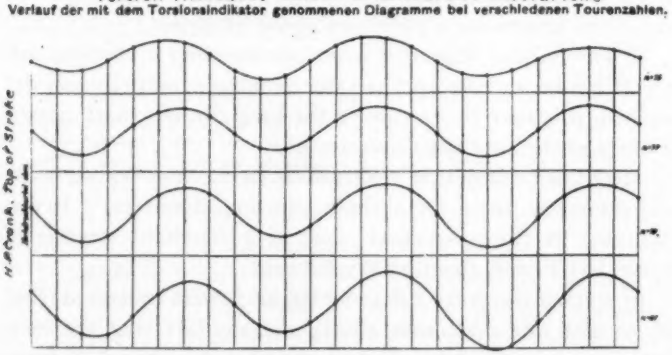

Torsion Indicator Curves at Different Revolutions
Verlauf der mit dem Torsionsindikator genommenen Diagramme bei verschiedenen Tourenzahlen.

Fig. 42.

planimeter on the original diagrams, the mean torque and therefore the E.H.P. of the engine was obtained. The modulus of elasticity used was that ascertained by Frahm, 830,000 Kg/cm^2 (11,801,936 pounds per square inch).

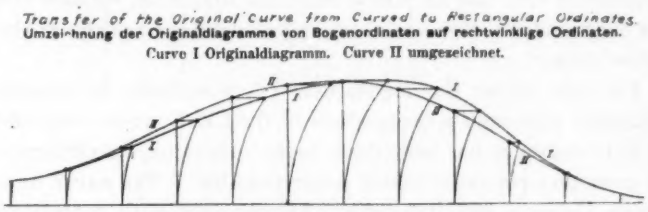


Fig. 43.

Since two sets of levers and pencils were set opposite each other, each recorded half the diagram and gave thus a good measure of their accuracy.

The horsepowers obtained vary, for the most part less than 1 per cent., and only at the lower powers as much as 2 per cent.

The developed curve of E.H.P. revolutions is shown in Fig. 44. The poor position of many of the points may be traced

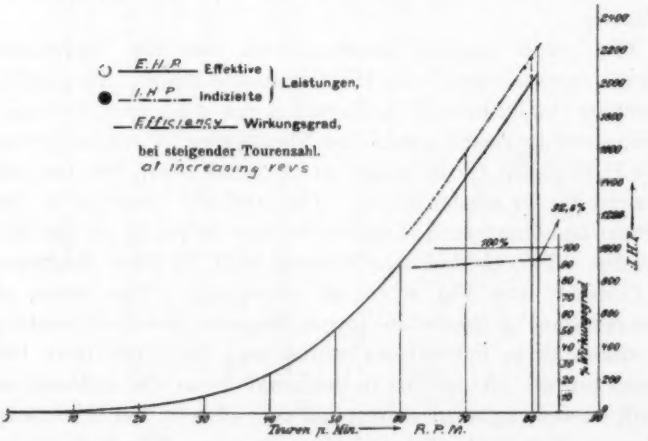


Fig. 44.

back to inaccurate counting of the revolutions with the help of a watch, something which is possible even on the largest merchant steamers for lack of a proper tachometer, as the revolutions are not constant. This disadvantage is especially evident in the case of steam indicator diagrams, because the six diagrams of a three-cylinder engine cannot be taken at the same instant.

For this reason the mechanical efficiency may be reliably obtained only from a comparison of the I.H.P. curve with the E.H.P. curve, as has been done here; otherwise, an efficiency of over 100 per cent. would seem possible. The curve here given shows a slow falling of the efficiency with decreasing load, the maximum being over 92 per cent. with 2,200 I.H.P. or 2,000 E.H.P.

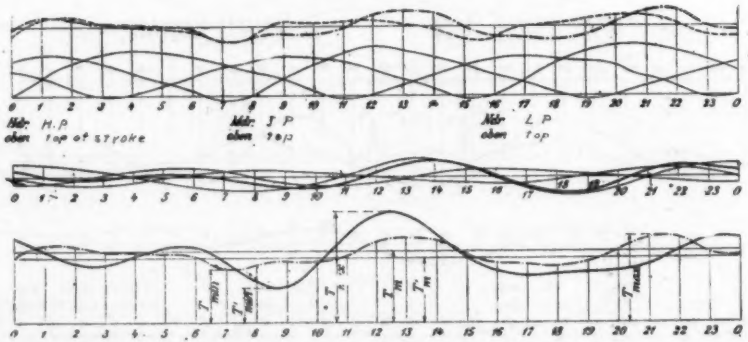
Of greatest interest would be a comparison of the torsion-indicator curves with their corresponding torque curves.

In Figs. 45 to 48 are shown, for four different revolutions, the torque due to steam pressures alone and then with the added moments of inertia of the reciprocating parts. For the higher revolutions 77 and 81 the moments of inertia were obtained corresponding to the variable experimental peripheral velocity, which varied but little from the mean peripheral velocity.

The lowest diagram shows in each case the theoretical torque curve alongside the effective torque curve. To exactly compare the latter with the former is not at all easy, for when the recording pencil passes the "beginning of stroke" point the H.P. piston is no longer at the dead point, but has advanced by the torsion angle. The author's ideas as to the proper combination of these curves may be found in the Appendix. This method was followed with all these diagrams.

Consider now Fig. 45 for 66 revolutions. The course of the resultant or theoretical torque diagrams is very favorable; it shows three fluctuations which vary but little from the mean torque. According to the usual views the ordinate of both curves ought to correspond very closely, but a cursory glance will show that this is seldom the case. The fluctuations

Comparison of the experimental with the theoretical torque curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve
$n = 66$ Umdrehungen. *Revs*



Verlauf der Tangentialdrücke für:	*Course of the theoretical curve for*
Dampfdrücke allein.	*Steam press. only*
Dampfdrücke mit Massendrücken der oscill. Massen	*Steam press. with moments of inertia of the oscillating masses.*
Resultierendes Massendruckdrehmoment (für mittlere Drehgeschwindigkeit).	*Resultant turning moment of inertia (for mean speed)*
Resultierende Tangentialkraftkurve.	*Resultant theoretical curve.*
Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet)	*Experimental torque curve. (made by the torsion indicator)*

Theoretical curve:
Tangentialkraftkurve:

$$\frac{T_{max}}{T_m} = 1{,}22$$

$$\frac{T_{max}}{T_{min}} = 1{,}63$$

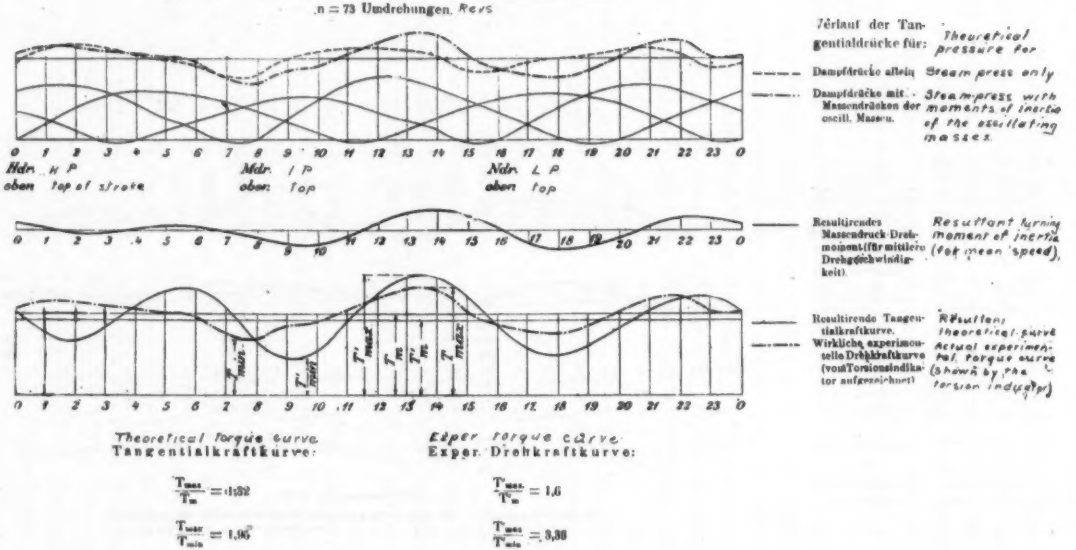
Experimental curve:
Experimentelle Drehkraftkurve:

$$\frac{T_{max}}{T_m} = 1{,}69$$

$$\frac{T_{max}}{T_{min}} = 3{,}15$$

Fig. 45.

Comparison of the experimental with the theoretical torque curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve.


n = 73 Umdrehungen. Revs
Verlauf der Tangentialdrücke für: Theoretical pressure for
Dampfdrücke allein Steam press only
Dampfdrücke mit Massendrücken der oscill. Massen Steam press with moments of inertia of the oscillating masses
Mdr. H P oben top of stroke
Mdr. I P oben top
Mdr. L P oben top
Resultierendes Massendruck-Drehmoment (für mittlere Drehgeschwindigkeit) Resultant turning moment of inertia (for mean speed)
Resultirende Tangentialkraftkurve. Resultant theoretical curve
Wirkliche experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet) Actual experimental torque curve (shown by the torsion indicator)
T max
T min
T m

Theoretical torque curve.
Tangentialkraftkurve:

$\frac{T_{max}}{T_m} = 1{,}32$

$\frac{T_{max}}{T_{min}} = 1{,}95$

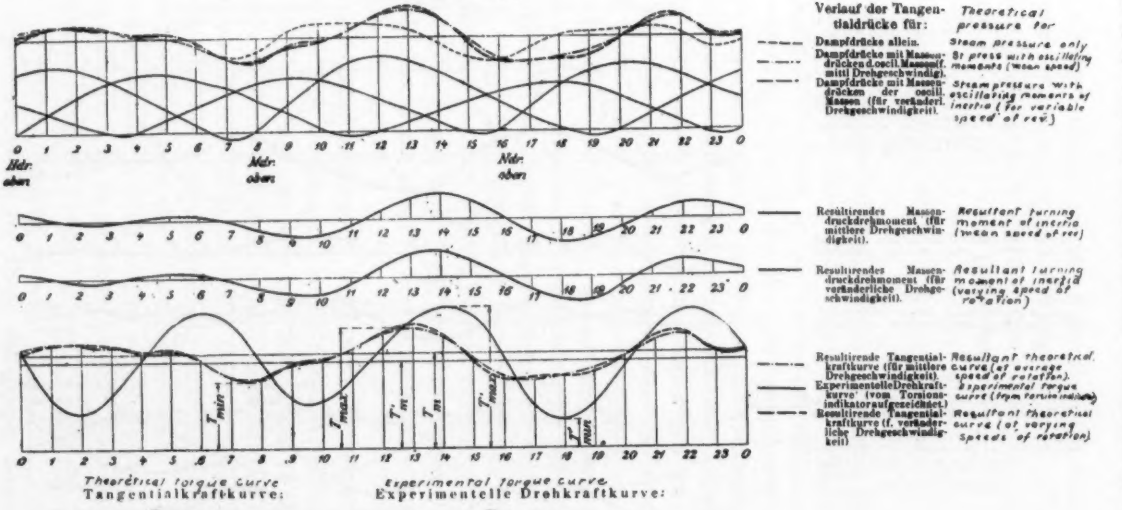
Exper. torque curve.
Exper. Drehkraftkurve:

$\frac{T_{max}}{T_m} = 1{,}6$

$\frac{T_{max}}{T_{min}} = 3{,}38$

Fig. 46.

Comparison of experimental with theoretical curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkurve.
n = 77 Umdrehungen. Revs.

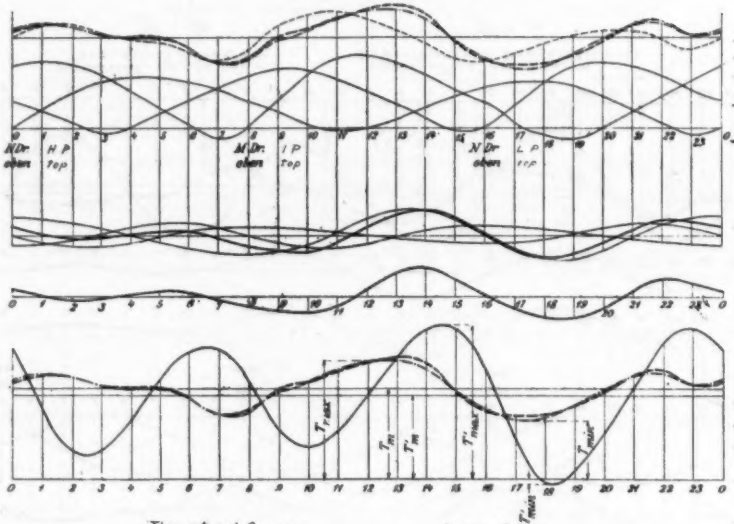

Verlauf der Tangentialdrücke für:
Theoretical pressure for
Dampfdrücke allein.
Steam pressure only
Dampfdrücke mit Massendrücken d. oscill. Massen (f. mittl. Drehgeschwindig.).
St. press. with oscillating moments (mean speed)
Dampfdrücke mit Massendrücken der oscill. Massen (für veränderl. Drehgeschwindigkeit).
Steam pressure with oscillating moments of inertia (for variable speed of rev.)
Mdr. oben
Resultirender Massendruckdrehmoment (für mittlere Drehgeschwindigkeit).
Resultant turning moment of inertia (mean speed of rev.)
Resultirender Massendruckdrehmoment (für veränderliche Drehgeschwindigkeit).
Resultant turning moment of inertia (varying speed of rotation)
Resultirende Tangentialkraftkurve (für mittlere Drehgeschwindigkeit).
Resultant theoretical curve (at average speed of rotation).
Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet).
Experimental torque curve (from torsionindicator)
Resultirende Tangentialkraftkurve (f. veränderliche Drehgeschwindigkeit)
Resultant theoretical curve (at varying speeds of rotation)
T_{max}
T_m
T_{min}
Theoretical torque curve
Tangentialkraftkurve:
Experimental torque curve.
Experimentelle Drehkraftkurve:

$\frac{T_{max}}{T_m} = 1{,}27$ $\qquad$ $\frac{T'_{max}}{T'_m} = 1{,}62$

$\frac{T_{max}}{T_{min}} = 1{,}79$ $\qquad$ $\frac{T'_{max}}{T'_{min}} = 5{,}0$

Fig. 47.

Comparison of the experimental with the theoretical curve.
Vergleich der experimentellen Drehkraftkurve mit der Tangentialkraftkurve.
n = 81 Umdrehungen. Revs



Theoretical Pressure for
Verlauf der Tangentialdrücke für:

- Dampfdrücke allein. *Steam pressure only*
- Dampfdrücke m. Massendrücken d. oscill. Massen (f. mittl. Drehgeschwindigkeit)
- Dampfdrücke m. Massendrücken d. oscill. Massen (f. veränderl. Drehgeschwindigkeit)
- *Steam press. with moment of inertia of the reciprocating masses (at average speed)*
- *Steam pressure with moment of inertia of reciprocating masses (at variable speed)*
- Resultirendes Massendruckdrehmoment (für mittlere Drehgeschwindigkeit) *Resultant turning moment of inertia (at mean speed)*
- Resultirendes Massendruckdrehmoment (f. veränderliche Drehgeschwindigkeit) *Resultant turning moment of inertia (at variable speed)*
- Resultirende Tangentialkraftkurve (für mittlere Drehgeschwindigkeit.)
- Experimentelle Drehkraftkurve (vom Torsionsindikator aufgezeichnet.)
- Resultirende Tangentialkraftkurve (für veränderliche Drehgeschwindigkeit)
- *Resultant theoretical curve (at average rotating speed)*
- *Experimental torque curve (shown by torsion indicator)*
- *Resultant theoretical curve (at varying speed of rotation)*

Theoretical Curve
Tangentialkraftkurve

$\frac{T_{max}}{T_m} = 1.30$

$\frac{T'_{max}}{T'_{min}} = 2.0$

Exper Curve
Experimentelle Drehkraftkurve

$\frac{T_{max}}{T_m} = 1.85$

$\frac{T'_{max}}{T'_{min}} = -20.4$

Fig. 48.

of the effective torque are much greater than those of the theoretical torque and are partially opposed to it. The failure of the two to coincide is shown by the ratio:

$T\,max. : T\,min. = 1.22 \times$ theoretical torque;

$T'\,max. : T'\,min. = 1.69 \times$ effective torque.

Still more different are the values of $T\,max. : T\,min. =$ 1.63 to 3.15.

Fig. 46, for 73 revolutions, shows still greater difference. The lowest diagram shows the comparison. Following the two distinct fluctuations in the theoretical torque curve we might expect the same for the effective torque curve; but instead we have three distinct oscillations of the shaft, of well rounded form and without any minor variations. The fluctuations of the effective torque are here also much greater than might be expected, the ratios being $T\,max. : T\,min. = 1.32$ and 1.6, and $T\,max. : T\,min. = 1.95$ and 3.33.

Fig. 47, for 77 revolutions, shows still greater torsional oscillations, whose maximum amplitude is 1.6 times the mean torsion, and the theoretical diagram shows the ratio, $T\,max. : T\,min. = 1.27$. Very noticeable is the difference of the opposite ratios of $T\,max. : T\,min.$, namely 5.0 and 1.79. Striking is the fact, shown in Fig. 46, for 73 revolutions, that the first oscillation of the effective torque is exactly opposite to the theoretical torque.

The comparison shown by Fig. 48, for 81 revolutions, is surprising. The two distinct oscillations of the theoretical torque ought to call forth only two fluctuations in the effective torque; but instead there appear three strong torsional oscillations per revolution, of which the last one actually assumes the nature of a negative turning moment, which would show that for an instant the advancing propeller dragged the combined shaft and moving engine weights around with it. It is, therefore, not to be wondered at that there appear the very different ratios of $T\,max. : T\,min. = 1.30$ and 1.85 and $T\,max. : T\,min. = 2.0$ and -29.4.

The only explanation for these differences occurring at various revolutions lies in the "Torsional Oscillations of the System."

A calculation of the oscillations of the shaft was made from this formula:

$$N = \frac{30}{\pi R}\sqrt{\frac{G\,\theta\,(M + m)}{L \cdot M \cdot m}},$$

where

R = crank radius,

G = modulus of elasticity in kg/cm^2,

θ = polar moment of inertia of the shaft's cross section in cm^4,

L = shaft length reduced to cm,

M = propeller mass, } reduced to the crank circle.

m = engine mass, }

The result was 243 oscillations per minute, with 81 revolutions. $243 \div 81 = 3$ torsional vibrations of the shaft exactly for one revolution, which would show 81 revolutions to be a "critical" speed. Indeed, the oscillations might be as high as 250, and the close agreement with the experiments is very good, as Figs. 42 and 48 show.

We may conclude further that with 61 revolutions we may expect $243 \div 61 = 4$ oscillations. The experiment proves it, for with 63 revolutions we had four distinct oscillations.

It may be further expected that with intermediate revolutions we will have a combination of the two frequencies of oscillation. Figs. 41 and 42 show this very plainly. The gradual change from a simple curve to the large oscillations at 81 revolutions demonstrates plainly the correctness of the calculated "critical" revolutions.

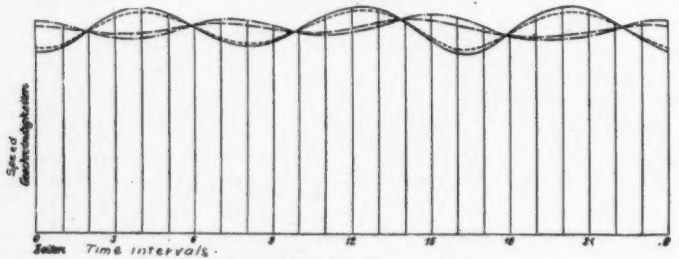
We might justly object to these assertions that the steam indicator would give exactly the same curves if its cylinder were made to oscillate with varying speed of the shaft.

Now, a rough calculation of the oscillations of the indicator will show that they are often several times as great in number as those of the shaft.

In order to remove the least objection and to test the accuracy of the indicator the torsion of a 27.5-meter shaft was measured by electric sparks with one or two inductors.

The torsion curves earlier obtained showed for correspond-

Speed curves
Verlauf der Geschwindigkeitskurven,
$n = 78$ Umdrehungen. Revs.



Speed curves for
Geschwindigkeitskurven für:

Forward drum **Vordere Messtrommel** } *experimentally obtained*
After drum **Hintere Messtrommel** } **experimentell ermittelt.**

Engine **Maschine** } *obtained graphically*
Propeller } **graphisch ermittelt.**

% of dissimilarity
Ungleichförmigkeitsgrade:

Engine **Maschine:** $d = \frac{v_{max} - v_{min}}{v_m} = \frac{1}{4{,}4} = 22{,}5\ \%$

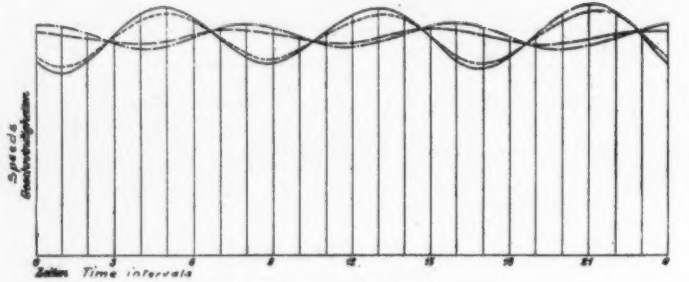
Forward drum **Vordere Messtrommel:** $= \frac{1}{5{,}2} = 19{,}1\ \%$

After drum **Hintere Messtrommel:** $= \frac{1}{11{,}9} = 8{,}4\ \%$

Propeller **Propeller:** $= \frac{1}{8{,}5} = 11{,}8\ \%$.

Fig. 49.

Speed Curves
Verlauf der Geschwindigkeitskurven
n = 81 Umdrehungen. *Revs.*



Speed Curves for:
Geschwindigkeitskurven für:

Forward drum
Vordere Messtrommel } experimentell ermittelt. *obtained experimentally*
After drum
Hintere Messtrommel

Engine
Maschine } graphisch ermittelt. *obtained graphically*
Propeller

% of dissimilarity:
Ungleichförmigkeitsgrade:

Engine **Maschine:** $\delta = \frac{n_{max} - n_{min}}{n_m} = \frac{1}{8{,}16} = 31{,}6\,\%$

Forward drum **Vordere Messtrommel:** $= \frac{1}{3{,}93} = 25{,}5\,\%$

After drum **Hintere Messtrommel:** $= \frac{1}{13{,}1} = 7{,}6\,\%$

Propeller **Propeller:** $= \frac{1}{8{,}7} = 11{,}5\,\%$

Fig. 50.

ing revolutions the same general form as the indicator diagram. To obtain a quantitative comparison of the results the difference $T\,max. - T\,min.$ was obtained from different diagrams and the percentage of their variation calculated for both methods of measurement. Out of nine such calculations there was a mean difference of 1.2 per cent., which is regarded as sufficiently accurate for most physical measurements. Small inaccuracies in the vicinity of the zero line were eliminated by this difference method.

At the same time the varying speeds of the propeller and the engine were obtained from the spark diagrams of both drums by a method described in the appendix. Figs. 49 and 50 show the diagrams for 78 and 80 revolutions. The lack of uniformity varies for the different parts of the shaft, and at 80 revolutions that of the engine attains the high value of 31.6 per cent.

RESULTS AND CONCLUSIONS.

Having described the experiments, let us now briefly consider the results and the conclusions that may be drawn.

I. Most important is the fact that in the torsion indicator we have a technical apparatus which automatically, and without any mental effort, shows the effective torque in the form of a diagram, through the integration of which we obtain the E.H.P. Of great significance is the fact that shaft lengths of only 1 to 2 meters are necessary, while up to this time Frahm's method of torsion measurement required from 20 to 30 meters length.

Torsion measurement is therefore now made possible with ships, especially warships, whose shafting is short.

Another point is that all parts function perfectly, and the difficulties of the methods considered may be regarded as solved.

II. The close agreement of the tests made with entirely different methods shows the perfect reliability and accuracy of the diagrams. If we properly inform ourselves as to the moduli of elasticity of the shaft materials by means of experi-

mental shafts, we open a long perspective of practical as well as scientific measurements.

We are then in a position to ascertain from a single diagram the E.H.P. transmitted to the propeller, while, for example, 16 diagrams are necessary in the case of an eight-cylindered pair of steam engines.

This fact is of importance in connection with steam turbines, which we have no other way of indicating. The modification shown in Fig. 37 permits of direct reading of the torque from a scale.

The mechanical efficiencies of marine engines may also be simply obtained, and thus one of the most uncertain factors in the design of a new ship and of a propeller may be eliminated.

Obviously, it would be advantageous to science to profit by the easily-made torsion measurements, especially if, by the methods previously described, the variations of the revolution speeds be registered in the same drum. The actual speed of any point of the shaft may then be obtained from the single combined diagram. In the appendix it is further shown how to find the course of the frictional power and its overcoming in the engine. At the same time might be determined the energy absorbed by the ship's vibrations, a method for which may be found in the appendix.

III. It might be the case for special reasons that the first experiments with the apparatus would give quantitative results, but, of course, the test would have to be subordinated to the program of the trip.

In close agreement with the experiments of Frahm are the results here obtained, and they go to show the untenableness of the old theory that the actual torque is closely proportional to the theoretical torque.

Frahm's experiments were principally conducted either at the normal or the critical revolution speeds. The systematic tests described in the foregoing pages, and made at all usual speeds, prove the existence of torsional vibrations at speeds quite different from the critical speeds, and show the course

of the vibrations for a wide range of revolution speeds, something which is of the greatest importance in practical application.

The question is now, naturally, asked, how may these results be practically applied to the calculation of shaft dimensions?

It would be very wrong to wish to shake in the slightest the usual methods of design, namely, those of the various bureaus of standards, for they represent the decade's experience in all normal cases. The fact that the actual diagrams do not agree with the theoretical at all revolutions need not disturb us in the least, for this is taken in account, in a large measure, by the coefficients of the bureaus of standards or classifications. Very seldom have inexact coefficients been dangerous to the engineer, for it is the practice to copy former designs. Dangerous only are the authoritative view points which make insufficient allowance for abnormal cases, such as that where the normal revolution speed happens to be also the critical.

In order to avoid this it is possible, as Mr. Frahm has pointed out, to separate the critical from the normal speeds by changing the diameter of the shaft.

With reference to the practical execution of Mr. Frahm's proposal I should like to add the following:

1. If it is possible to calculate in advance the oscillations of the shaft only after having ascertained its elastic properties from exact model tests—that is, the torsion of the cranks, thrust shafts, flanges, couplings and shaft accessories—then primarily only a rough calculation is possible for the reduced shaft length, so that the calculation for the rate of oscillation could be only approximate.

2. Let it be emphasized that it is possible to change the calculated rate of oscillation in ships with long line shafting if the line shaft is weakened or strengthened, and that in this case it would be better not to change the diameter of the propeller and crank shafting.

It is of the greatest practical interest to ask how, in the

case of finished ships, existing dangerous torsional oscillations may be decreased in order to prevent a broken shaft.

A change in the rotating engine masses by the addition of a flywheel is seldom possible, and, even so, the rate of oscillations or vibration would not thereby be much affected, for in the formula used the engine masses appear under the radical in both the numerator and denominator.

In practice an easy way is to vary the work done in each cylinder by adjusting the link gear until the vibrations are reduced. The form of the theoretical diagram is thereby changed, as are therefore also its harmonic waves, which alone maintain the dangerous vibrations and determine their amplitude. We are indeed here restricted to rough trials; meanwhile, in my tests, I watched with safety for the possibility, even in a small degree, of dangerous vibrations.

A change in the harmonic waves set up and an eventual decrease in their amplitude may be obtained by changing the crank-angles, although that would be much more expensive than changing the positions of the links.

Finally, the dangerous vibrations may be overcome by changing the revolutions per minute. The tests described, for example, show a considerable decrease in the vibrations at the normal speed of 75 revolutions, as compared with those observed, the (in this case) exceptional speed of 81 revolutions. Of course, a change in the revolutions is often only slightly, or not at all, available, and then perhaps associated with many other difficulties.

IV. In conclusion, I wish to refer to the strains observed in the lathe experiments, to which we may compare the strains caused by jars and jolts in the propeller- and crank-shafts.

It now only remains for me to thank here the management of the Vulcan Engine Company, Stettin, through whose cooperation was made possible the development of the apparatus and the carrying out of the experiments. My thanks are also due to the graduate engineer, Mr. Jahn, who took part of the diagrams after my directions.

It would be highly desirable if my experiments were ex-

tended to warships, for the torsional vibrations measured at all higher revolutions seem to give, in most cases, a factor more or less useful in showing the needs of long marine-engine shafts.

APPENDIX.

COMBINATION OF THE EFFECTIVE TORQUE CURVE WITH THE THEORETICAL TORQUE CURVE.

To combine the effective torque curve given by the torsion indicator with the theoretical curve we must first reduce them to the same base.

Obviously, the ordinates of both must be reduced to the same pressure scale. The mark on the recording drum corresponding to top of H.P. stroke is good only for a torsionless shaft; during work the H.P. crank has passed the dead point by the torsion angle at the instant the indicator pencil marks the dead point. This holds good for every crank position.

Following is a method of combining the diagrams:

Both diagrams are divided into the same number, m, equal parts. Neither these division points on the two diagrams nor their ordinates correspond in point of time. One division represents an angle of 360 degrees $\div m$. The torsion angle may easily be obtained in degrees from the length of the torsion arc. If, for example, for the nth division point of the diagram there is an angle of i degrees, then the corresponding point on the theoretical diagram is i degrees further on. In this way we obtain a new division of the theoretical diagram, of which every division point corresponds exactly with one on the torsion diagram. More difficult is the division of the theoretical diagram into equal time intervals, for which it is necessary to construct the integral curve of the speed diagram.

GRAPHIC METHOD OF DETERMINING THE SPEEDS OF THE PROPELLER AND THE ENGINE.

Available are the experimental speed curves of the after and the forward recording drums, best laid off with time intervals.

In Fig. 51 there are drawn four vertical parallel lines

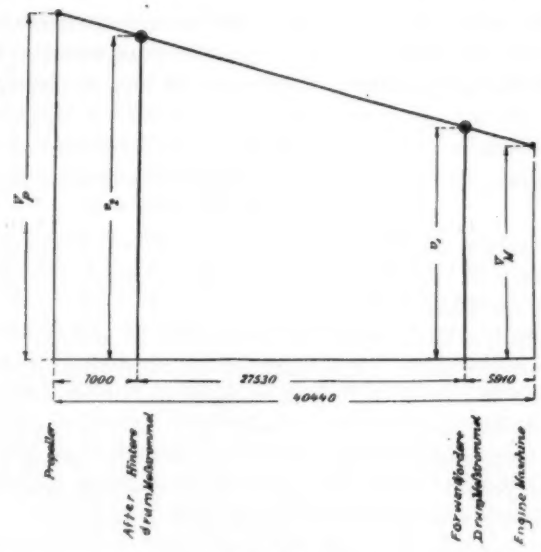


Fig. 51.

whose distances apart are proportional to their relative position on the shaft reduced to a uniform diameter. On any convenient scale are laid off V_1 and V_2, the simultaneous speeds measured on the two drums. A line produced through these two points cuts on the other two perpendiculars the desired speeds V_m and V_p.

The proof is as follows:

Four points which lie in a straight line on a torsionless shaft lie in a spiral line when the shaft is doing work, and this spiral line when unwound is an inclined straight line. This graphic construction shows the resultant unwinding at two different instants, the beginning and the end of the interval Δt. If in this time the two inside points have traveled the distances V_1 and V_2, so must the outer points have gone through the distances V_m and V_p. If $\Delta t = 1$, then V_1, V_2, V_m and V_p give the speed directly.

The base line need not be perpendicular to the parallel lines.

The peripheral speeds of any other shaft points may be found by drawing in the proper parallels. This construction gives the same results as those analytically obtained by Frahm.

DETERMINATION OF THE VARIABLE SPEED OF THE SHAFT BY THE TORSION INDICATOR.

This may be obtained experimentally as follows, from the normal torsion diagram.

The removable recording drum is insulated by a wooden or other base from the ship's hull and connected with the + pole of the secondary of an induction coil, whose — pole is connected with the shaft, and, thereby, with the torsion indicator and its pencils. If the primary circuit is broken at equal intervals the sparks in the secondary will pierce the diagram paper. Large distinct holes are obtained by placing a Leyden jar in parallel with the secondary circuit. In order to confine the sparks to one recording pencil the other pencils are insulated by hard-rubber strips.

The arc between two holes measured on a proper scale gives the speed at any instant for the corresponding part of the shaft.

The operation may be carried out successfully with the usual type of interrupter, but careful speed measurements require a mercury turbo-interrupter or some other motor type.

From the single combined diagram the speed for each shaft point may be found from the following considerations:

The measured speed of any one point of the shaft would be a measure for that of any other point if the shaft's torsion did not vary during the measuring interval.

In reality the speed of point No. 2 diverges from the measured absolute speed of point No. 1, while the relative speeds of points 1 and 2 come together, which is caused by a variation in the torsion arc.

The torsion indicator furnishes for the measured shaft length the multiplied torsion arc, and this is proportional to the torsion of any other shaft length.

The unknown relative speed may then be easily found by subtracting from each other the torsions at the beginning and

the end of the time interval and applying the proportional part of this difference to the shaft length between points 1 and 2.

DAMPING IN THE ENGINE.

The method is analogous to Frahm's determination of the coefficients of propeller resistance.

The action of the theoretical and of the effective torque is given, and the constructions above described show the variations in the speed.

Next, by differentiation, from the speed curve there is constructed a curve showing the accelerations of the speed. On another scale the acceleration curve shows the variation in the forces which are absorbed or given off by the rotating and oscillating masses at any change in the speed. If we add these acceleration forces to the effective torque applied to the shaft and subtract this sum from the theoretical torque we have in the remainder the forces used up in friction.

The difference measuring suggested assumes that all the values taken have been accurately determined.

Further discussion on this subject I will reserve for some later time.

DETERMINATION OF THE ENERGY ABSORBED BY HULL VIBRATIONS.

This is based on the fact that a considerable absorption of energy by hull vibrations occurs only at critical revolution speeds, where the energy absorbed is proportional to the square of the amplitude of the vibrations.

If the vibrations occur in the engine, the following method may be used.

All the shafting, including the thrust shaft, is uncoupled, and a test is made at no load with gradually increasing revolutions. As in Fig. 52, a curve is made for the horsepowers at no load and their corresponding revolutions. In this case the number of revolutions sometimes even exceeds the normal.

At critical speeds there enters a greater or less irregularity in the performance curve, somewhat like that in Fig. 52. An

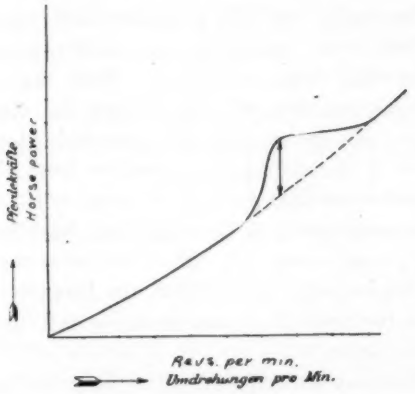


Fig. 52.

increase in the work done is here accompanied by scarcely any rise in revolutions, until the critical revolution speed is passed and exactly the opposite is true—a sudden increase in revolutions with a very small increase in work. If we prolong the curve we can pick off directly the horsepower used in generating the hull vibrations. If we measure at the same time the absolute vibrations we may obtain the damping constants for the vibrations.

This phenomenon was observed long ago in the case of bicycling. There are certain kinds of paving which, in connection with pneumatic tires, produce such vibrations that the speed can be increased only after large increase in power. The rider, indeed, finds this waste of energy highly inconvenient.

If the hull vibrations are caused by the propeller, not only will the I.H.P. be higher than the revolutions require, but the E.H.P. will be higher also. The irregularity of the curve is not here so noticeable because the power here absorbed by the vibrations is only a very small percentage of the whole E.H.P.

Considerable difficulty exists if the vibrations are due to both the engine and the propeller. It is quite possibly the

case that the engine and the propeller each cause strong vibrations, which often neutralize each other, giving the ship complete freedom from vibration. This depends, as Mr. Gumbel, Hamburg, first pointed out, on the size of the resultant forces, as may be observed from their phases.

A solution of the difficulty is possible from the fact that efficiency determinations go hand in hand with accurate pallographic measurements of the amplitude between the limits of vibration in each case. If the vibration is the same when the engine works with and without the propeller, the energy actually absorbed may be found as above described by reducing the horsepower.

If the vibrations are different after the propeller is coupled up, we are reduced to model experiments.

Further discussion of this subject would exceed the scope of the present article.

DESCRIPTION AND TEST OF 27-INCH CURTIS TURBINE FOR 50-FOOT U. S. NAVY CUTTER.

By Ensign Paul E. Dampman, U. S. Navy, Member.

This turbine, intended for use in a 50-foot U. S. Navy Cutter, was built at the works of the Fore River Ship Building Company at Quincy, Mass. The contract requirements were that the installation of machinery intended for the cutter should develop 250 brake horsepower, as determined by test in the works of the company. After considerable delay, the turbine was completed on September 1, 1907, and received its official trial on September 11 and 12, 1907, at the works of the company.

DESCRIPTION OF TURBINE.

The turbine is of the usual Curtis type, with four ahead stages and one reverse stage. The pitch diameter of the wheels is 27 inches. The first ahead wheel and the reverse wheel each have four rows of rotating blades, while the remaining wheels have three rows of blades each.

Steam is admitted to the first ahead stage from the steam chest through seven nozzles. Of these, three are always open, the remaining four being provided with closing valves operated by means of a socket wrench. The astern steam chest has eight nozzles, none of which have closing valves. The stuffing glands on the shaft are packed with carbon rings. The weight of the turbine complete is 4,100 pounds. The designed horsepower is 250 when making 1,000 revolutions per minute, and using dry steam at 250 pounds gauge.

Two views of the turbine are shown. One shows the tur-

bine in the power house, complete for shipment; the other shows the details of rotor wheel and blading. The method of blading is interesting. A steel rod, of channel section, is punched with holes, spaced the distance apart of the blades. The blades, of extruded bronze, are driven into the punched holes and the shank riveted over. A series of saw cuts are then made in the under sides of the channel bar, enabling it to be bent to the curvature of the rotor wheel very much as a piece of molding is bent into a curve. The sides of the channel bar are then caulked into slots on the rotor, and the whole strengthened by means of a steel shroud ring, riveted over.

METHOD OF TESTING.

The turbine was mounted in the power house upon a cast-iron bedplate. Beneath this bedplate was a Wheeler surface condenser of 490 square feet of cooling surface. After condensation, the steam was delivered to a pair of measuring tanks, mounted on platform scales, by an Edwards single air pump of 10 inches stroke, 5-inch steam cylinder and 12-inch air cylinder.

The turbine shaft was directly connected to a water brake, having an arm 5.252 feet in radius. Thus, brake pull, in pounds, times r.pm.. divided by 1,000 is equal to brake horsepower. The brake pull was measured by means of a platform scale.

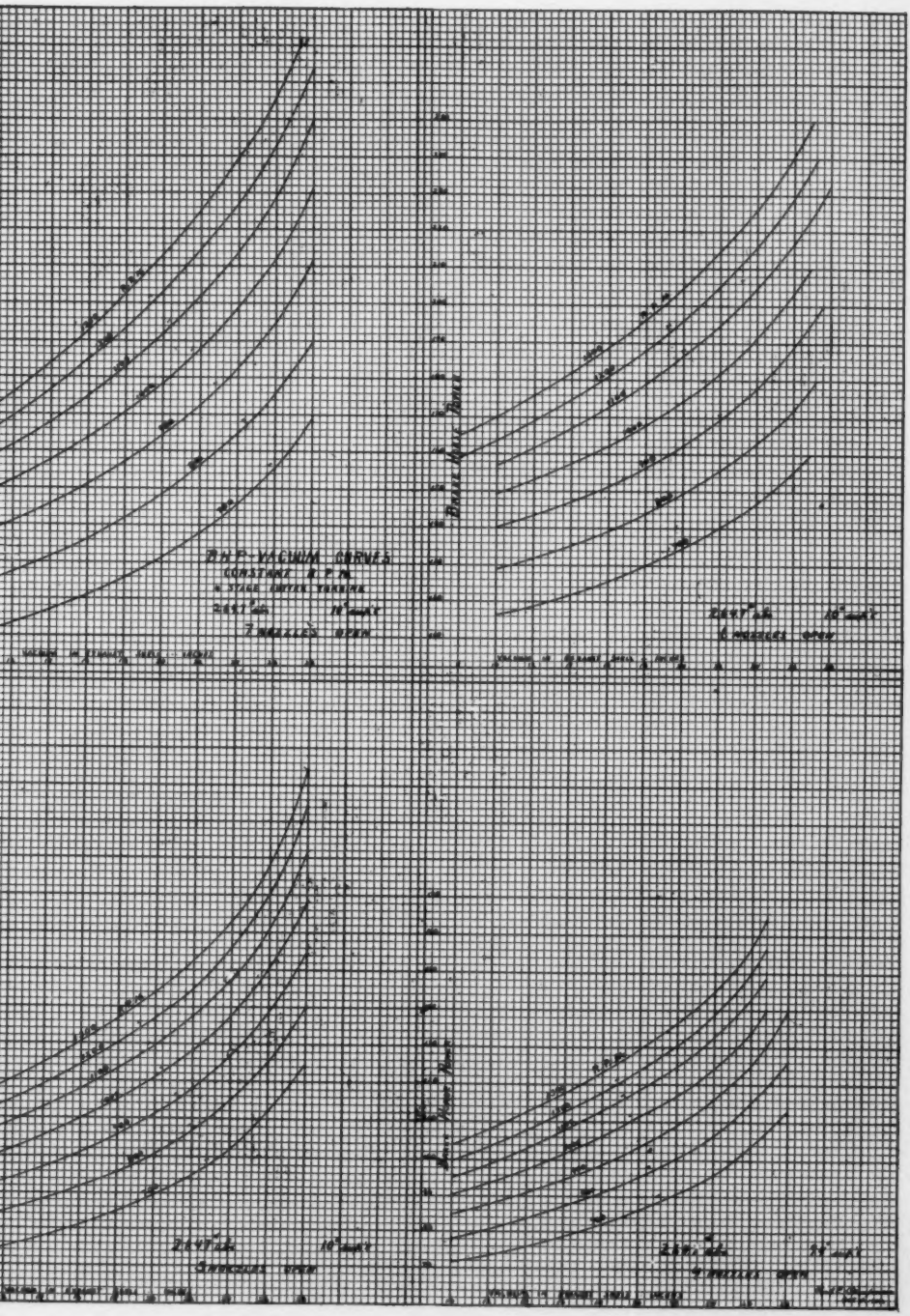
Steam was supplied for the test by a Babcock & Wilcox boiler, forming one of the Company's power plant, of the following dimensions:

Grate surface, 43 square feet; heating surface, 1,956 square feet.

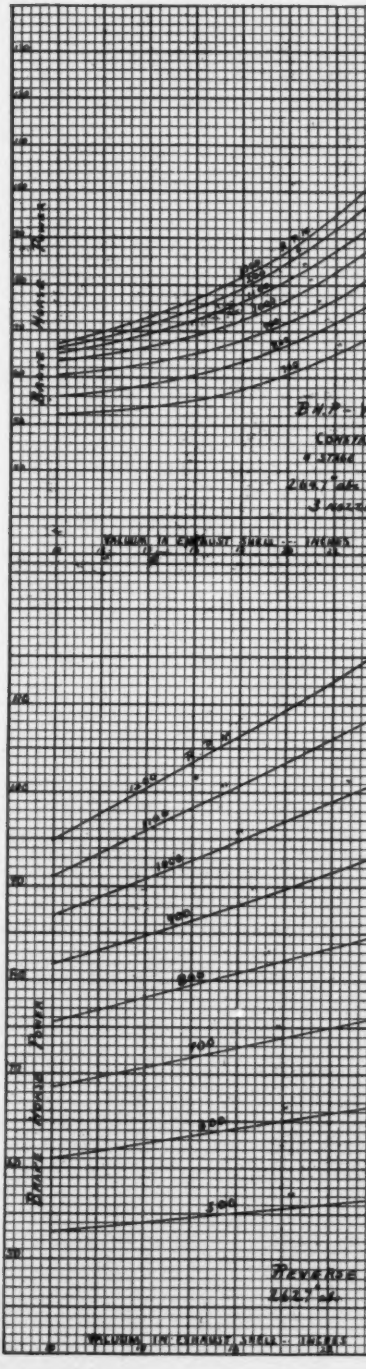
This boiler was provided with a superheater, but only a part of the steam used was passed through the superheater. The object of having the steam slightly superheated was to be sure that the steam entering the turbine was dry and saturated.

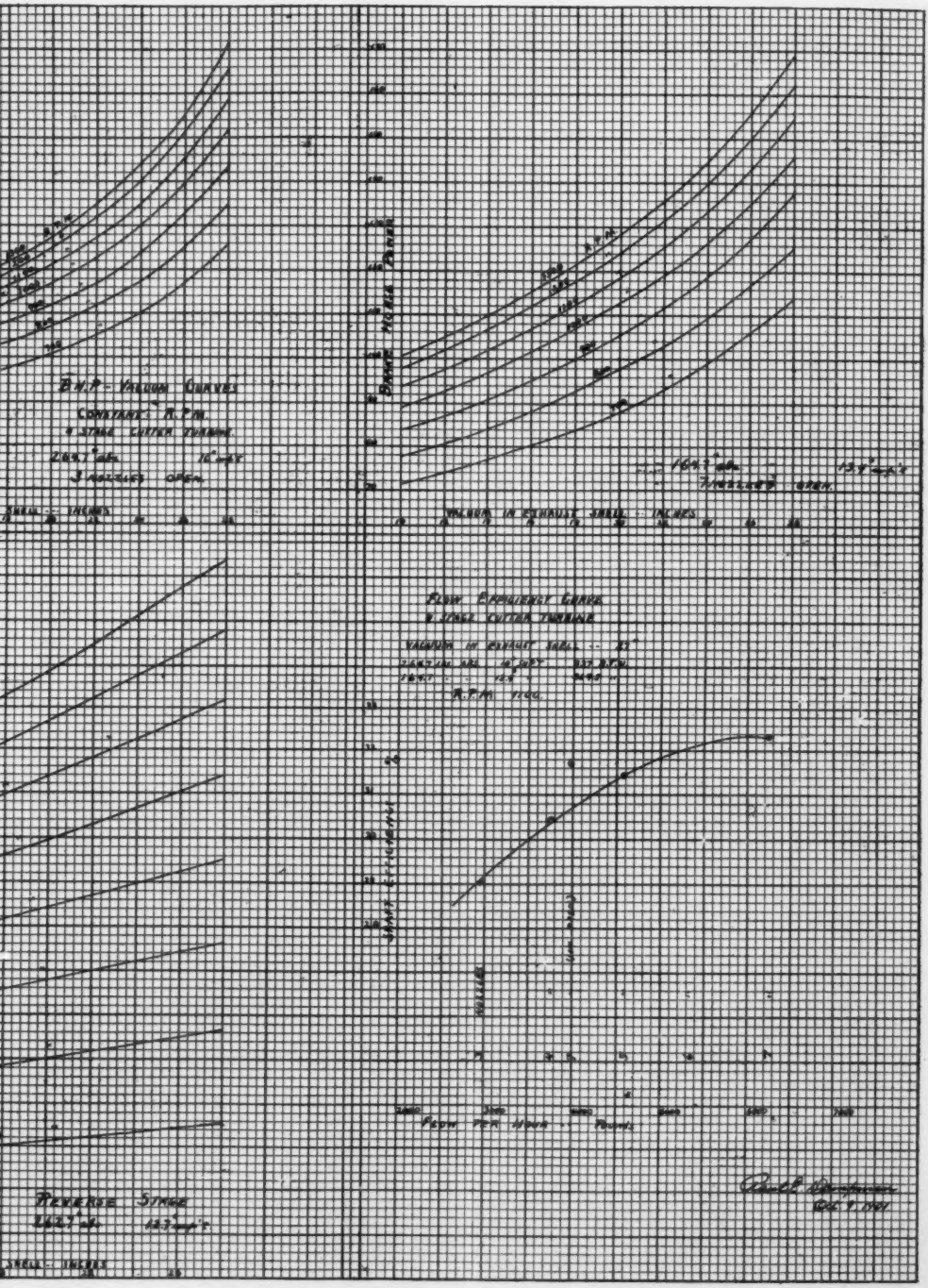
In the separator, just before the steam reached the turbine,

XU

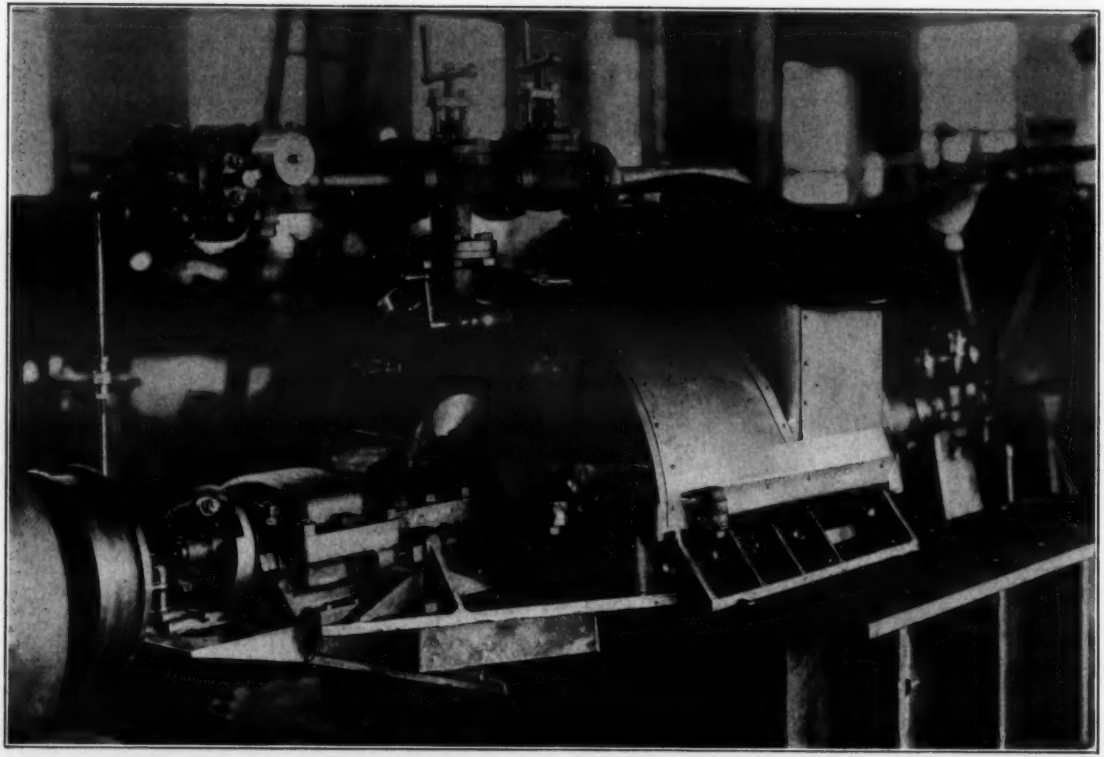


Vacuum Curves, 27-Inch Curtis Turbine.

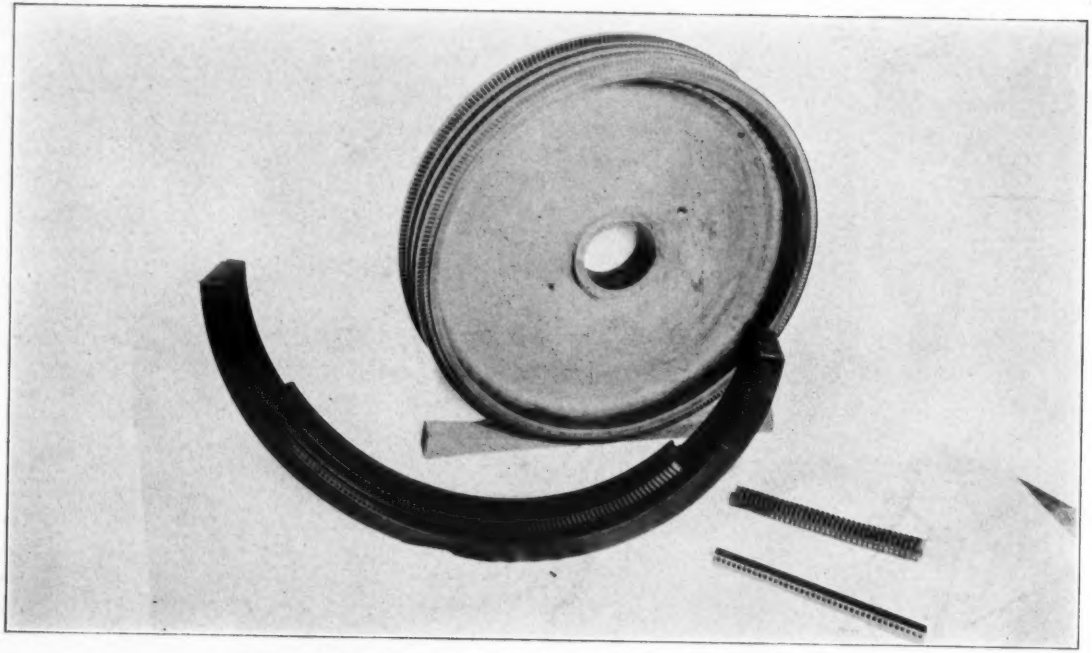




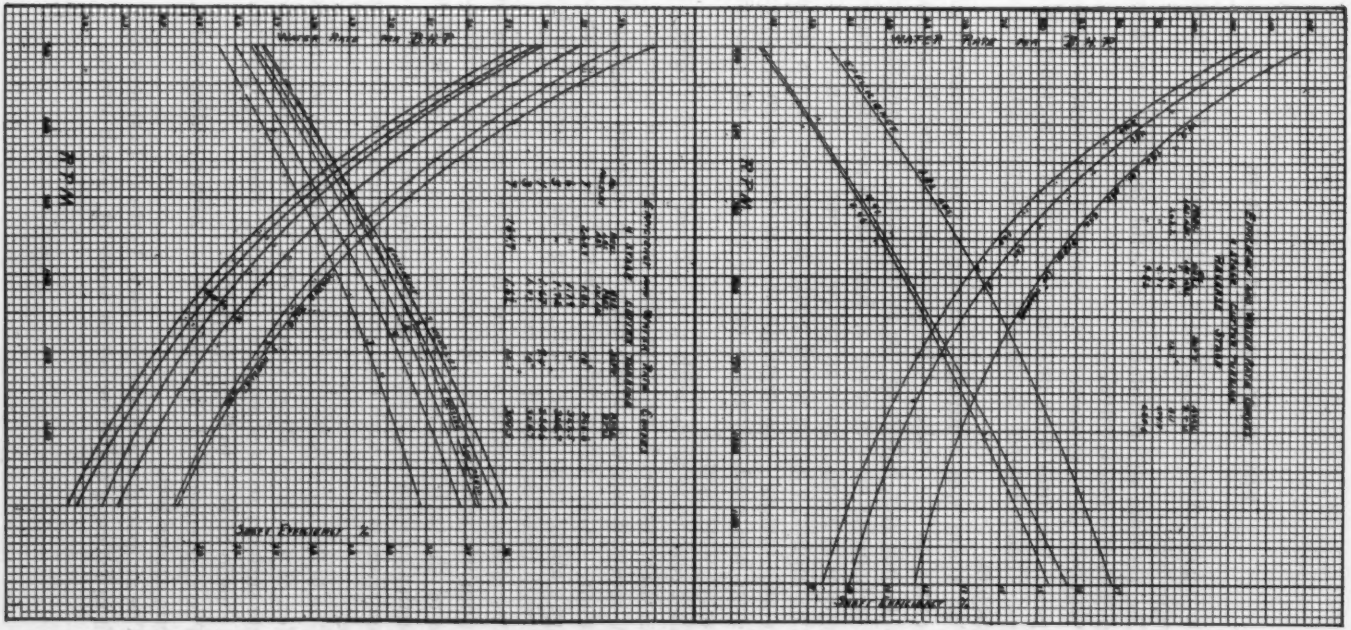
Vacuum Curves, 27-Inch Curtis Turbine.



27 Inch Curtis Turbine, Complete.



DETAILS, ROTOR WHEEL AND BLADING, 27-INCH CURTIS TURBINE.



EFFICIENCY AND WATER RATE CURVES, 27-INCH CURTIS TURBINE.

a cooling coil was inserted to give better control of the degree of superheat.

The pressure was maintained constant in the steam chest by means of a globe valve.

Pressures were measured by means of calibrated steam gauges. Vacuums were measured by means of mercury columns. Temperatures by means of high-grade glass thermometers.

The speed was controlled by regulating the amount of water fed to the brake. The speed was indicated by means of a tachometer, belted to the main shaft. The readings of the tachometer were checked at intervals of several minutes, by means of a small counter, thrown in and out by hand.

The highest vacuum which could be obtained was a little over 26½ inches. To regulate the vacuum, when a lower vacuum was used, air cocks on the condenser and exhaust chamber were opened.

Test runs were made with 3, 4, 5, 6 and 7 nozzles open, at three different speeds and four different degrees of vacuum.

A reversed test was made with three different degrees of vacuum. Also, using reduced steam pressure (150 pounds gauge), ahead tests were made, using the full seven nozzles and with three degrees of vacuum.

The object of making all these tests at different degrees of vacuum was so that some idea might be had of the performance of the turbine after it was installed in the launch, with a particular view to comparing the performance of the turbine-driven launch with engine-driven launches of the same type.

The boiler constructed for use in the launch is a water-tube boiler, of the Fore River type, of the following dimensions: Grate surface, 16.62 square feet; heating surface, 555 square feet; ratio of H.S. to G.S., 33.39 to 1.

The condenser is of the surface type, with a cooling surface of 203 square feet.

The air pump is a Blake Simplex Vertical Featherweight, with cylinders 4 and 8 inches in diameter, 6-inch stroke.

DATA SHEET—TRIAL OF 27-INCH CURTIS TURBINE.

Nozzles open.	Average steam flow per hour, pounds.	Exhaust vacuum, inches.	Chest pressure, pounds, gauge.	Chest temperature, Fahrenheit.	Pressure, first stage, pounds, gauge.	Pressure, second stage, pounds, gauge.	Vacuum, third stage inches.	R.P.M.	B.H.P.	R.P.M.	B.H.P.	R.P.M.	B.H.P.
Ahead 7	6,255	26.3	250	416	66.5	11.8	14.7	1,096	260.9	1,188	274.7	1,283	287.6
	6,255	24.0	250	416	66.5	11.7	14.5	1,091	244.2	1,185	256.9	1,282	269.2
	6,255	20.0	250	416	66.5	11.8	14.6	1,095	220.7	1,187	232.7	1,280	241.8
	6,255	15.0	250	416	65.0	11.3	11.15	1,092	200.6	1,191	210.1	1,284	216.2
Ahead 6	5,329	26.35	250	416	53.0	7.7	16.9	1,095	219.5	1,185	231.1	1,284	240.0
	5,329	24.0	250	416	53.0	7.3	16.9	1,093	200.8	1,189	213.2	1,280	218.4
	5,329	20.0	250	416	53.0	8.3	15.75	1,095	185.4	1,187	194.9	1,282	202.8
	5,329	15.0	250	416	53.0	8.2	12.0	1,095	169.4	1,188	176.4	1,281	182.2
Ahead 5	4,576	26.5	250	416	43.0	5.0	18.3	1,098	185.4	1,196	195.5	1,284	201.5
	4,576	24.0	250	416	42.5	4.8	18.4	1,094	166.7	1,192	175.7	1,283	180.0
	4,576	20.0	250	416	42.5	4.7	16.75	1,095	148.9	1,188	156.9	1,281	162.7
	4,576	15.0	250	416	42.7	4.8	12.6	1,093	134.4	1,186	141.8	1,279	145.6
	4,576	10.0	250	416	42.7	4.9	8.15	1,091	123.8	1,186	128.0	1,282	134.0
Ahead 4	3,748	26.6	250	415.4	33.0	11.8	20.2	991	138.5	1,149	152.7	1,281	161.9
	3,748	24.0	250	415.4	32.7	11.5	20.25	991	123.3	1,147	136.3	1,281	143.2
	3,748	20.0	250	415.4	32.7	11.5	17.7	993	110.9	1,150	121.3	1,281	129.0
	3,748	15.0	250	415.4	32.7	11.6	13.35	994	100.2	1,146	107.2	1,279	113.4
	3,748	10.0	250	415.4	32.7	11.6	8.75	993	90.3	1,149	96.9	1,279	103.5
Ahead 3	2,916	26.5	250	416	23.0	4.3	22.5	702	79.1	992	101.9	1,279	113.9
	2,916	24.0	250	416	22.35	4.2	21.65	702	70.1	992	89.1	1,280	103.4
	2,916	15.0	250	416	22.5	4.3	14.0	702	54.5	993	68.1	1,279	74.4
Ahead 7	3,915	26.7	150	378	34.0	7.0	20.0	993	138.7	1,150	151.1	1,279	159.4
	3,915	20.0	150	378	34.0	6.9	17.65	994	110.2	1,148	121.8	1,280	127.6
	3,915	10.0	150	378	34.0	7.0	8.6	992	88.9	1,148	95.1	1,280	100.0
Reverse. All nozzles (8)	6,072	25.0	247.5	418	...	...	...	799	85.4	995	102.1	1,191	116.4
	6,072	20.0	247.5	418	...	...	...	796	81.9	994	97.0	1,187	108.5
	6,072	10.0	247.5	418	...	...	...	797	75.4	993	86.3	1,188	94.7

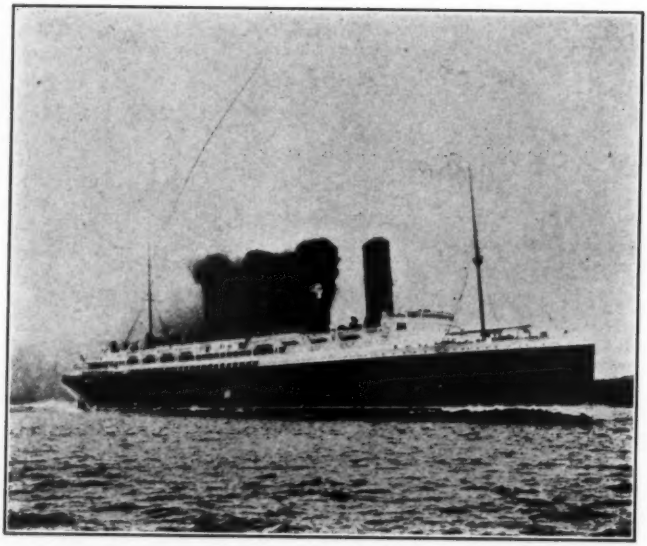
The feed pump is of the vertical, single-piston type, with cylinders 4 inches and 2½ inches diameter, and 4-inch stroke.

The circulating pump consists of a 10-inch impeller driven by a 2¼-inch by 2-inch single engine.

Following are the weights of the machinery:

	Pounds.
Turbine,	4,100
Shafting,	587
Boiler,	5,262
Condenser,	1,138
Air pump,	280
Feed pump,	173
Circulating pump,	111
Total,	11,651

This does not include piping and connections, or water. Allowing 600 pounds for the necessary piping, and 2,000 pounds for water, the weight of machinery per designed B.H.P. comes out approximately 57 pounds.



THE CUNARD EXPRESS STEAMER "LUSITANIA."

THE CUNARD LINER *LUSITANIA.*

In her official trials, carried out under the direction of the technical staff of the Cunard Company and of the Admiralty representatives, the *Lusitania* met the most sanguine anticipations of all concerned. At a draught of 30 feet, she has steamed over 26 knots on the measured mile; on a 48 hours' sea run on long measured distances she has maintained a mean speed of 25.4 knots. The contract anticipated a speed of 24½ knots on the round voyage on the Atlantic.

The uprecedented length of the trial precluded "jockeying." The course of about 300 miles was traversed four times in alternate directions, so as to eliminate the influence of tide and weather. And thus any speed maintained on such a trial may be continued indefinitely, so long as coal and other supplies are available. It is unnecessary to say that the machinery worked satisfactorily. The general result stated carries conviction from this point of view. Before entering upon this, the most crucial test, the *Lusitania* had made several preliminary trials on the Clyde measured mile, not only to tune up the turbine machinery but to standardize the relation between revolutions, power and speed, so that a series of trials could be made to determine the coal consumption at various speeds.

For the economy tests the vessel was loaded to a draught of 32 feet 9 inches, equal to a displacement of 37,000 tons. Water and coal consumptions were taken while the vessel ran for six hours at speeds of 15, 18 and 21 knots respectively. The results were thoroughly satisfactory, but the data obtained were in connection with service requirements rather than scientific purposes. On the run to the Firth of Clyde, the starting point of the full sea-speed trials, the course measured out on the chart was between the Corsewall Light on the Wigtownshire

Coast to the Longship Lighthouse at Lands End, and this had to be traversed four times, alternately north and south. The compass bearings gave the distance, which aggregated about 1,200 miles. The weather was favorable, with cloudless days and starlight nights; but on both nights north-west winds freshened to forces of six and eight, and although this occurred when the vessel was steaming north, and somewhat increased resistance and slightly reduced speed, it brought consolation in the fact that it prevented fog. The feature of the trial was the uniformity of the speed on both runs south and on the two runs north, the latter being against the wind and tide. The course was divided into three approximately equal parts by the Fodling and Tuskar lights. Compass bearings taken at these intermediate points proved the uniform rate of steaming. The time taken on the runs south differed by only two minutes; further proof is unnecessary of the great regularity of steam supply or of turbine efficiency. The speed on four runs was: South, from Corsewall, 26.4 knots; north, from Longship, 24.3 knots; south, from Corsewall, 26.3 knots; north, from Longship, 24.6 knots; mean speed, 25.4 knots. We omit second-placed decimals, but, in any case, the percentage of error in observation is, with such distance, negligible.

This is a great performance: it exceeds by two nautical miles per hour any similarly long run made. The truest significance lies in the uninterrupted mechanical precision with which every unit of the machinery worked. The air pressure in the ash-pits of the boilers did not at any time reach the maximum of ¾ inch prescribed in the specification by the Cunard Company. The boiler pressure averaged 186 pounds per square inch, while the pressure at the receiver of the high-pressure turbines varied little from 150 pounds; at the low-pressure receiver it was 3½ pounds. The mean vacuum was 28.2 inches, with an average barometric reading of 29.8 inches. The mean revolutions of the four shafts were 188 per minute, and the power, according to the torsionmeter, was 64,600 horsepower. To those not versed in the details of steam-turbine performances the fact is illuminative. The circumferential or tip velocity of the ro-

tors of the low-pressure turbines was 150 feet per second, equal to over 9,000 feet per minute. The general procedure in the machinery department accorded with Atlantic practice, and the performance might to all intents and purposes have been two days' running, each equal to over 600 miles, on a voyage to New York. On returning to the Clyde the vessel proceeded on shorter distance tests betwen Corsewall and Chicken Rock, the southern extremity of the Isle of Man, and between the Holy Isle and Ailsa Craig, in the Firth, while following this were progressive runs on the measured mile at Skelmorlie, also on the Clyde; these ranged up to 26 knots, as already indicated in our introductory sentence.

The steering qualities of the vessel have also been tested. When steaming at 15 knots the rudder was put from amidships to hard over, both to port and starboard, in 15 seconds, and the full circle was completed in 8 minutes. Immediately before commencing to turn the engines were running at the rate of revolution which gave 15 knots. A careful record of revolutions was made on a time basis during the evolution, and it was found at the completion of the circle that the rate of revolution was then 70 per cent. of the rate at 15 knots. The final speed was thus assumed as 10.5 knots, the average speed 13 knots, and the diameter of the circle about 1,100 yards. This for a ship of this great length is a most satisfactory performance; the ship, at 22 knots, made the complete circle in 7½ minutes, with 15 degrees of helm. In ordinary steering the ship answered her helm very rapidly, according to the testimony of the pilot, and her swing was easily checked. Although the water was very fine, alike on the 36-hours' run around Ireland and on the 48-hours' trial on the deep-sea course, there was sufficient swell on the Atlantic in the first-named trip, and between the Tuskar and the Longship Lights on the subsequent runs, to cause pitching and rolling motions to be perceptible, and to afford opportunity for repeated records. In respect to these points a large number of observations gave the period of a single roll from side to side as almost exactly 10 seconds; a single pitch occupied 4 seconds. This

latter result calls for no remark, and the slow rolling indicates that, while the vessel has a satisfactory measure of stability, the long period of roll assures us that there will be the minimum of discomfort to passengers through this cause. The trials were finished August 1st.

As to vibration, it has for years been the custom at the Clydebank Works to make observations with Mr. Otto Schlick's pallograph, and the instrument has been used on every trial of the *Lusitania*. Mr. Schlick enunciated, in a paper read some years ago at the Institution of Naval Architects, a formula for determining approximately for various types of ships the first period of vibration and the slowest period of vibration of which the structure as a whole is capable. The correctness of this formula was borne out by the observations made on the *Lusitania*. The amplitude of vibration, although very small, was quite distinctly marked, and at a draught of 32 feet 6 inches and a displacement of nearly 37,000 tons, a first period vibration of a frequency of 62 per minute was clearly recognized on the pallograph records..

GOVERNMENT AND CUNARD LINE AGREEMENT; MAIL AND WAR-SERVICE SUBSIDY.

Under this agreement the Government provided a sum sufficient to pay for the new vessels, not exceeding £2,600,000, secured on debentures at 2¾ per cent. interest, while in addition £150,000 was to be paid per annum, on condition that the company would cause to be built, in the United Kingdom, two steamships of large size, capable of maintaining the minimum average ocean speed of 24½ knots in moderate weather. In the event of this speed not being maintained, and if the speed does not fall below 23½ knots a deduction is to be made from this annual payment by agreement. The clause in which this speed condition is set out is a matter of very considerable interest, and may here be quoted:

"If in the case of either of the two steamships mentioned in Clause 3 hereof, or any vessel substituted therefor, the company shall, before such steamship sails on her first voyage, fail

to adduce to the satisfaction of the Admiralty reasonable proof from trials that such vessels will be capable of maintaining a minimum average ocean speed of 24½ knots an hour in moderate weather, but shall prove to the like satisfaction that such vessel will be capable of maintaining an average ocean speed of not less than 23½ knots an hour under such conditions as aforesaid, then such deduction shall be made from the annual payment of £150,000 to be made by His Majesty's Government under the last preceding clause hereof as shall be agreed upon, or, failing such agreement, shall be determined by arbitration, by an arbitrator appointed by the Lord Chief Justice for the time being, and the decision of such arbitrator shall be final."

The minimum speed seems thus to be 23½ knots, although even then the ships may not be rejected; but the Cunard Company aim at, and will probably get, 25 knots, costly as that may be in respect of first charge, coal consumption, and upkeep. These two ships, in addition to carrying the mails and maintaining the prestige of Britain—which we regard as a very important commercial asset—are to be at the service of the Government in the event of war. To some it may seem remarkable that such an agreement should be necessary to secure the services of such vessels in emergency, and we may even have the naval critic urging that our cruisers ought to be equal in speed for any duty that the proposed Cunarders may fulfil. But many such forget the enormous difference between an Atlantic liner and a cruiser. In the first place such modern liners have a displacement twice that of the greatest cruiser built. In the merchant ship there is no armor to provide for, there are no guns to take, no ammunition nor naval stores, and the consequence is that the architect can allow for machinery something approaching double the weight per unit of full power. It follows that the same reliability in long-distance full-speed steaming cannot be guaranteed in cruisers. In other words, speed and weight of machinery are the main considerations in the merchant ship, whereas in the cruiser they are important, but probably equal only to gun power and armor protection. The Cunard ships cannot be equal to a cruiser in the latter

qualities for warfare; but they will be superior to similar ships which in time of stress can be withdrawn from the merchant service of other countries, and utilized for naval work.

It may be contended that patriotism alone ought to ensure that ships flying the British flag should ever be at the country's call; but the old order, which necessitated British ownership before the Union Jack could be flown, passed away in the adoption of the Limited Liability Act. Where the owning company is registered in this country under this Act the fact alone enables their ships to fly the British flag, even though not a penny is British capital, and not one shareholder—not even a director—is a British subject. This is not the place to enter into the question as to whether such a condition should be permissible. The British flag is certainly a great advantage to any ship in view of the maintenance of our naval power, and it is a question as to whether a change should not be made.

But the Cunard agreement with the Government involves the maintenance of the company as a purely British concern, one of the clauses of the agreement laying it down that the company will be under British control, "that no foreigner shall be qualified to hold office as a director of the company or to be employed as one of the principal officers of the company; and no shares of the company shall be held by, or in trust for, or be in any way under the control of any foreigner or foreign corporation, or any corporation under foreign control." The Cunard Company altered their Articles of Association to meet these conditions. While carrying on business to the best advantage, the company agree not to raise the freights or charges for the carriage of goods in any of their services; while no undue preference against British subjects is to be given. Plans of all ships to be built to steam 17 knots or upwards are to be submitted for approval; facilities are to be given for periodical inspection by the Admiralty, and for storing guns, ammunition, &c., at the ports; no chartering, except to the Indian Government, is to take place without notice being given to the Government, and the option to similarly charter. The Gov-

ernment is always to have the right of hiring the boats, the rates for such being: for vessels over 22 knots, 25 s. per gross register ton per month, and 5s. more if the company provide officers and crew; and for 20 to 22 knots, 20s., and 4s. more for staff; and for slower boats at less rates. In addition to holding the ships at the service of the Government, it has been prescribed in the agreement that all the officers, and three-fourths of the crew shall be British subjects, and that a large proportion shall belong to the Royal Naval Reserve. The ships are thus to be utilized as a great training school for British officers and seamen, and each month a record is to be made of the personnel with this point in view.

As to the mail service, the company shall, during the term of this agreement, convey, by means of mail ships from Liverpool (viâ Queenstown), or from Queenstown to New York, once in every week, on such day as may be provided, all such mails as shall for the purpose of such conveyance be tendered or delivered at Liverpool and Queenstown respectively, and shall employ as mail ships the fastest of the steamships for the time being belonging to or chartered by the company. For this mail service there shall be payable to the company a yearly sum after the rate of £68,000 per annum; but whenever in any one week more than 100 tons measurement (that is to say, 4,000 cubic feet) of parcel mails (exclusive of empty receptacles) in the aggregate are conveyed in either direction (whether by the mail ships or by any other steamships of the company), a further sum of 26s. 3d. is to be paid for every complete ton measurement; but the Postmaster General may, at his option, pay the rates of freight for the time being charged by the company on similar parcels to other companies or firms whose business is to carry parcels; but all parcels for which the said rates of freights are paid by the Postmaster General shall be carried by the company, subject to terms and conditions similar to those upon which the parcels of such other companies or firms are carried, and not under the terms and conditions of the agreement.

The loan of 2½ millions for the building of the two ships

is secured by a charge upon the whole of the company's assets, including all vessels built for or acquired by the company, so long as such steamships, or vessels, or any of them, shall remain the property of the company. The loan is to be repaid by the company by annual instalments, each of which shall be equal to one-twentieth of the total amount of the advance, the term of the agreement being for twenty years.

Under the trust deed under which the loan or debentures are to be issued three trustees hold office, the Government and Cunard Company's nominees electing the third. The company are further to issue to two nominees of His Majesty's Government one £20 share of the company, carrying the same voting power and other rights and privileges as an ordinary £20 share of the company; but, for the purpose of demanding a poll in respect of, and voting against, any special resolution involving any alteration of the company's articles of association, so far as respects the provisions referred to in the agreement, also carrying the following additional rights and privileges—namely, the right to demand a poll upon the occasion of any such special resolution, and the right to give against any such special resolution additional votes equal in number to one-fourth of the number of votes possessed by the company's share, stock or debenture holders for the time being.

THE CONTRACT; THE DIMENSIONS OF THE "LUSITANIA."

While the order for one of the two ships (*Lusitania* and *Mauretania*) was placed with Messrs. John Brown & Co., Limited, the other was awarded to Messrs. C. S. Swan, Hunter and Wigham Richardson, of Wallsend-on-Tyne, the machinery for the latter vessel being ordered from the Wallsend Slipway and Engineering Company, Limited.

The dimensions of the *Lusitania* are: Length over all, 785 feet; length between perpendiculars, 760 feet; breadth, molded, 88 feet; depth, molded, 60 feet 4½ inches; gross tonnage, 32,500 tons; draught, 33 feet 6 inches; displacement, 38,000 tons; number of passengers—first cabin, 552; second cabin, 460; third cabin, 1,186; type of engine, Parsons turbine; number and type of boilers, twenty-five cylindrical; number of furnaces,

192; steam pressure, 195 pounds; total heating surface, 158,350 square feet; total grate area, 4,048 square feet; draught, Howden's; total indicated horsepower (designed, 68,000 speed (designed), 25 knots.

One important feature dealt with in fixing the designs had reference to the use of the ships as cruisers or scouts in time of war, the machinery—which is almost entirely under the water line—has been so disposed in separate compartments, and with coal protection along each side, as to counteract, as far as possible, the effect of the enemy's fire at the water line. For purposes of attack the *Lusitania* will be provided with an armament as satisfactory as the armored cruisers of the County class, because on one of the topmost decks there will be carried, within the shelter of the heavy shell plating, four 6-inch quick-firing guns attaining a muzzle energy of over 5,000 foot-tons, while on the promenade deck on each side there will be four more guns on central pivot mountings, also able to penetrate 4¾-inch armor at 5,000 yards range, and 6-inch armor at 3,000 yards range. With the great speed, which can be maintained for three or four times the period that any modern cruiser can steam even at only 21 knots, and with the careful subdivision for protection and their satisfactory offensive power, the *Lusitania* and her consort may be regarded as most effective additions to any fighting squadron. Their advent is therefore a great advantage from the point of view of British sea power.

The rudder and steering gear are all placed well below the water line. This is a most important point in respect of protection, should these vessels be ever impressed into the national service. The stern has been suitably shaped in the *Lusitania* to enable this object to be accomplished satisfactorily.

THE ADOPTION OF STEAM TURBINES.

The second problem in design was the question of the type of propelling machinery to be adopted. The power of the machinery, for the dimensions and form evolved in the Government tank at Haslar, was 68,000 indicated horsepower. Ex-

perience has produced an exact rule as to the efficiency of tank boilers as steam generators, and thus the capacity of the boiler installation was a more or less fixed quantity; but it was important to determine whether the steam efficiency of the steam turbine or of the reciprocating engine was greater. Even in a ship where speed is the first desideratum it is incumbent upon the designer to aim at economy, alike in first cost, working expenses, and maintenance charges. Three years ago, when the decision had to be arrived at, there were comparatively few data even as to the steam consumption of turbines, and less as to their durability; but the Cunard Company, with that wisdom which has brought them to the front rank, decided to appoint a commission of experts to investigate and report upon the whole question. The company were equally felicitous in their selection of the experts.

This Commission entered upon their work with great care, conscious of the responsibility resting upon them; on their decision the success of the new ships largely depended. Admiral Oram, who, as Deputy Engineer-in-Chief of the Navy, has tackled the question of turbine economy, alike from the scientific and practical standpoint, with enormous advantage to the Navy, was able to put before the Commission very important results as to the performance of destroyers. The late Mr. Brock and Mr. Parsons assisted the Committee with the tests of Channel steamers into which their respective firms had fitted turbine engines. The resultant data, and a careful consideration of the performance of turbines ashore, encouraged the Committee to make the bold step of recommending for these huge liners the new prime mover. Events have since justified this intuition, and the consequence is that the Cunard liners are in the forefront, not only in speed, but in the method of attaining it. The constructors of the machinery, in accepting the contract for turbine machinery, with the heavy guarantees attached, also displayed characteristic enterprise.

The Clydebank firm entered upon their work in a thoroughly practical way, laying down immediately a complete turbine installation, where, by dynamo-metric means, they were able

to test the power, while at the same time measuring the water consumption. Many questions affecting the details of design were similarly experimentally determined, and the result was that they induced the Cunard Company to at once adopt turbine machinery in a large steamer of 20 knots speed, the order for which had just been placed with them. We have already dealt very fully with the machinery in this ship, the *Carmania,* and, at the same time, have described the experiments undertaken and the modifications made in the turbine, so that it is not necessary here to say more regarding this phase of the subject. It may, however, be indicated, as suggestive of the completeness of the experimental plant, that it was subsequently fitted to a Clyde steamer direct from the testing house, and has since given very satisfactory results, corroborative of the high efficiency attained at the works.

FORM OF STERN AND NUMBER OF PROPELLERS.

The adoption of turbines was immediately followed by very careful experiments as to the form of stern suitable for four propellers, and as to the proportions of propellers. Four screws were imperative, whether turbine or reciprocating engines had been adopted, because great ingenuity had to be devised in connection with the distribution of power in the cylinders of the piston engines immediately preceding Atlantic liners. In the *Campania* and *Lucania,* where the total power was 30,000 indicated horsepower, two sets of engines were found sufficient, each having three cranks with five cylinders, the high and low pressure being arranged tandemwise, with the intermediate cylinder in the center. In the next large ship, the *Kaiser Wilhelm der Grosse,* there were two sets of engines of the four-cylinder triple-expansion type, each cylinder working on a separate crank; although the power was the same, with a slight increase of steam pressure, the low-pressure cylinders were 96.4 inches in diameter. The Vulcan Company, in their succeeding ships, adopted various systems, and in the *Kaiser Wilhelm II* fitted four sets of engines, two on each shaft. Each set of engines had three cranks, and the tandem system

was again adopted for high and first-intermediate cylinders, but the diameter of the low-pressure cylinder was increased to 112.2 inches. This latter dimension has been exceeded only in one or two instances, in compound-paddle engines. The six cranks probably improved the uniformity of turning moment. With tandem cylinders the difficulties of balancing the moving parts were necessarily increased, and it is probable that, as a consequence, some portion of unnecessary weight had to be carried. In any case, the transmission of 20,000 indicated horsepower through a single shaft, setting up enormous torque, imposed a very severe condition even upon the best of steel makers, and those responsible for the design of the machinery in these new liners were, therefore, well advised in aiming at a reduction to about 16,000 or 17,000 horsepower through any one shaft.

The adoption of four units of machinery and four screw propellers enabled the machinery to be made in two completely separate sets, one to starboard and the other to port, just as in a vessel with twin-screw reciprocating machinery; a distinct advance upon the triple-screw arrangement in former turbine ships, where such complete independence is unattainable. The division of the power into two complete and independent systems follows the course pursued for so many years in the Royal Navy, whose lead is now so generally adopted in this respect in the mercantile marine. It has in this case enabled the engine rooms to be well subdivided by watertight bulkheads, and the advantages in general secured have been obtained without the sacrifice of a single point of any importance.

As in earlier Clydebank ships, the rudder, of the barn-door type, is supported for about two-thirds of its depth. Immediately forward of it, on each side, are the two inner propellers, the shafting for these being entirely borne within the ship, the framing of which was bossed out, and strongly supported by heavy webbing, as explained later. The forward propellers are about 70 feet ahead of the inside screws, and here also the frames are carried by heavy webs. Owing to the great beam of the ship and the very fine run, the blades of the outside propellers do not project beyond the beam line, while, at the same

time, all the propellers work in free water, and provision has been made for a satisfactory clearance between the propellers and the skin of the ship.

Another question affecting the efficiency of the propelling machinery had reference to the revolutions at which the propellers were to be run at full speed. In Channel-steamer work, and in small craft generally, a high rotating speed has been adopted, largely as a matter of necessity. Where weight is limited, it is important to minimize the dimensions of the turbine, and the velocity of steam being constant, the revolutions are of necessity high. The contention has been made—not always with due regard to experience—that the small high-speed propeller involves some loss in efficiency, especially in a seaway, and, further, detracts from the astern speed, and, indeed, from the general maneuvering power of the ship. Be that as it may, it was decided in the Cunard liners, after very careful consideration, to attain the full speed, with the propellers making about 140 revolutions per minute, and the turbines were proportioned to suit this speed. The peripheral speed being practically more or less constant, owing to the velocity of steam, and only affected by the angle or curvature of the blades, it became necessary to adopt turbines of very large diameter. Thus the rotor drum of the high-pressure turbines is 96 inches in diameter, and that of the low-pressure turbine 140 inches, the blades ranging from 2¼ inches to 22 inches in length. The result is to permit the use of a propeller of a diameter and pitch which will certainly remove any question as to relative efficiency, under normal conditions, as to maneuvering power and astern speed, and also as to the influence of head seas. From these points of view, the performance of the ship and the machinery will be watched with very careful interest; although the results already attained with the *Carmania* prove that the line of reasoning which has actuated the designers of the propelling machinery of the *Lusitania* is correct, and therefore there is every likelihood of a full practical success.

THE VENTILATION OF THE SHIP.

The thermo-tank system, as recently introduced by the Thermo-Tank Ventilating Company, of Glasgow, has been adopted in connection with the ventilation of the ship. This system aims especially at ensuring to all the living quarters of the ship a continuous supply of fresh air, which is not only warmed to the requisite degree, but is also humidified, so that none of the bad effects of over-drying can be felt. In cold weather the warmed air is discharged, through a regulated louvre, into each apartment, near the level of the ceiling; as it cools it gradually sinks to a lower level, carrying with it any carbonic-acid gas to the passage ways, where means are provided for allowing it to pass outside. In warm weather, or when heating is not necessary, the reverse action takes place, as the louvres near the ceiling constitute the exhaust, with the result that the warm impure gases leave the top of the room, and fresh atmospheric air comes in at the floor level.

The thermo-tank generally consists of an electric motor operating a fan which discharges air to the outside of a tube heater. The air then passes through the tubes, and comes in close contact with the heater surface, flowing thence to the main distributing trunks. Two valves are used for controlling the passage of air; one for regulating the temperature, while the mushroom valve on the top is provided for the exhaust air. It will be noted that the air passes round the outside of the heater on its way to the tubes, so that the loss from radiation is very small, the outer casing of the thermo-tank being quite cool on all occasions. The heater is warmed by steam from the main boilers, entering at the top, with an exhaust at the bottom. The pressure of steam is reduced to about 30 pounds, and a relief valve is fitted to blow off at from 80 pounds to 100 pounds pressure. The heater and all its connections are tested to the full boiler pressure. The air is humidified by means of a special valve admitting steam in a fine spray, by means of small needleholes in a copper hoop surrounding the heater. Tests carried out to compare the efficiency of the thermo-tank system with that of the ordinary heaters show that where the

steam-heated system took three hours to attain a given temperature, the thermo-tank only required fifteen minutes. The consumption of steam is small, as all the heat is abstracted, only water being drained off to the feed tank.

The first-class accommodation is connected to twenty-four thermo-tanks, which are arranged principally on the boat-deck houses. The second-class accommodation is connected to nine thermo-tanks, the third-class to eleven, and the officers' and crew's accommodation to five, these being arranged mostly on the top of deck houses, &c. The thermo-tanks in the fore end of the ship are placed between decks, and obtain their supply of fresh air from the after end of the navigating bridge, so that in this way a continuous supply of fresh air is ensured in the worst weather, there being no cowl heads or openings forward of the flying bridge. Although the thermo-tanks are arranged principally on the top of the boat-deck houses, the fresh-air supply is obtained from gratings opening out on the promenade-deck shelter. This has been done so as to avoid the smell from galleys, w.c.'s, &c., which all exhaust above the boat-deck houses. When the thermo-tanks are exhausting, of course, the cowl head provided for the purpose can then be used.

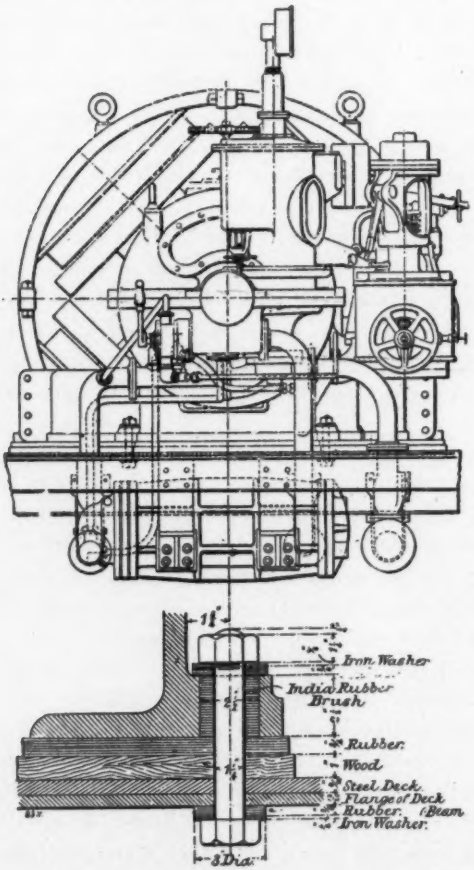
The thermo-tanks are capable of changing the air, either by exhaust or supply, in the various compartments to which they are connected at least from six to eight times per hour, and they are also capable of maintaining a temperature of at least 65 degrees F. in the coldest weather. In addition they are interconnected, so that in case of the breakdown of any thermo-tank, a supply can always be obtained from another.

The work of the thermo-tanks is further augmented by means of twelve powerful exhaust fans, which are connected by means of trunks respectively to all the galleys and pantries, bath rooms, lavatories and w.c.'s, these fans being of sufficient capacity to change the air in the above-mentioned compartments at least fifteen times per hour. In addition to the living quarters, the holds and other compartments, forward

and aft, are also mechanically ventilated, so that all natural ventilation requiring cowl heads, or openings through decks, has been dispensed with.

ELECTRIC LIGHTING.

An extensive electric generating station is arranged on a flat deck abaft the engine rooms. There are four generating



DETAILS OF TURRBO-GENERATORS.

sets, all alike, and each of 375 kilowatts capacity, the voltage being 110 to 120. The prime movers are Parsons turbines, and it is interesting to recall that the Clydebank firm was among the first, if not the first, to fit turbo-generators on ships, and on that occasion the vessel also was for the Atlantic trade. The turbines were designed to give full load when exhausting into a back pressure of 10 pounds. They run the dynamos at 1,200 revolutions per minute; but an overload of 10 per cent. for two hours is provided for. Each dynamo is shunt wound. The armature is of the surface drum-wound type, with one turn per section. No relative movement between the conductors and commutator bars is possible, but as an addi-

SUMMARY OF OFFICIAL TRIALS OF TURBO-GENERATORS.

Number of machine.	1034.			1077.	1078.			1079.
Date of test	4-8-06	4-8-06	4-8-06	14-8-06	13-8-06	13-8-06	13-8-06	14-8-06
Load	Full	Three-quarter	Half	Full	Full	Three-quarter	Half	Full
Stop-valve pressure	167	167	173	166	158	161	160	164
Barometer, in inches of mercury	29.77	29 77	29.77	29.54	29.4	29.4	29.4	29.54
Back pressure, in pounds	5.0	5.0	5.0	4.95	4.86	5.0	5.0	5.0
Speed, revolutions per minute	1,200	1,200	1,200	1,200	1,200	1,200	1,200	1,200
Voltage	111	111.2	113.6	115.2	107.0	109.8	112.3	114.5
Average kilowatts	373.27	288.9	188.42	375.38	371.75	285.28	188.3	373.31
Field volts	87 6	88.73	86 06	92 46	87.5	89 6	90.6	92 08
amperes	33.31	31.15	30.9	30.76	30.8	30.8	30.8	31.5
Average quantity of water per hour, in pounds,	17,831	15,017	11,419	17,301	17,888	15,104	11,649	17,546
Water consumption per kilowatt-hour, in pounds,	47.76	51 97	60.60	46.08	48.14	52.94	61.86	47.0

tional safeguard the connections between these are made flexible. Special driving horns in the ends of the core are provided. The insulation of the whole machine was tested with an alternating pressure of 2,000 volts between the conductors and the frame, and the insulation resistance after the above test was to be not less than one-half of a megohm for the whole machine, one megohm for the armature winding, one megohm for the field winding, and one megohm for each of the brush holders.

Tests were made of the various turbo-generators at the Heaton-on-Tyne Works of the constructors—Messrs. C. A. Parsons & Co., Limited—and the results for all four engines are given in the table. It will be noted that at half load the water consumption was in one case 60.60 pounds, and in

another 61.86 pounds per kilowatt-hour; at three-quarter load the consumption was 52 pounds to 53 pounds, and at full load from 46 pounds to 48 pounds, the back pressure in each case being about 5 pounds.

REFRIGERATING MACHINERY.

Two complete and independent installations of refrigerating machinery are fitted on board the ship, one for the preservation of the ship's provisions, and the other for the carriage of perishable cargo. Both have been constructed by the Liverpool Refrigeration Company, Limited, and must be described together. The ship's provision machine is situate near the forward end of the turbine-engine room on the main-deck level, and the chambers on the lower deck, port and starboard sides, some distance forward of the machine. These chambers have been insulated at the Clydebank Works with granulated cork in combination with specially-treated damp and rot-proof paper, with linings of white-pine boards. The chambers are divided into compartments for beef, mutton, poultry and game, bacon, milk, fruits and vegetables, and ice, and the wine and beer and spirit chambers are also lightly insulated and cooled to a suitable temperature.

The chambers have a total capacity inside insulation of about 13,000 cubic feet, and, in addition, there is a large cold larder on the upper deck, besides cold boxes in the first and second-class bars and still room. There is also an installation for the supply of cooled water for drinking and other purposes. In this connection every possible requirement has been thought out and arranged, not only for the preservation of the perishable provisions in bulk, but also for the convenience of the catering and culinary departments generally.

The installation is of the carbonic-anhydride type and consists of a horizontal compound duplex machine, mounted on a cast-iron box bed, which is divided by a longitudinal bulkhead into two portions, each of which contains an independent set of gas-condenser coils. These coils are of special soft-iron lap-welded tube, galvanized on the outside. The compound

steam cylinders drive from their tail-rods two horizontal double-acting CO_2 compressors. The crank shaft runs in four bearings, and is in two portions, coupled in the center and with a distance plate between the faces of the coupling. A neat arrangement of steam valves is fitted, so that the engine can work compound or independently as two high-pressure engines. By taking out the coupling bolts and distance plate each side of the machine can be run quite independently of the other. Cross-connections are provided, so that either compressor can deliver into either or both gas condensers, and the machine is the full equivalent of two independent machines combined on one base.

The evaporator is of the vertical type, the shell enclosing two independent nests of circular coils, one coupled to each compressor, with cross-connections, the same as for the condensers. Two horizontal duplex brass-fitted brine pumps circulate the brine, and a third, smaller and independent pump is provided for the special duty of pumping the brine supply to the cold larder and refrigerators in the saloon bars and other places independently of the main pumps. The whole brine distribution forms an entirely closed system. The brine is drawn from the evaporator and delivered into a distributing header, with valves and connections leading to the various pipe sections in the cold rooms. After passing through these the brine returns to a similar collecting header, and thence back to the evaporator, there being no open brine tank whatever. All the chambers are cooled with galvanized brine piping, arranged to suit various temperatures required in the several compartments, each of which is regulated independently of any other. The installation is, we believe, the largest and most complete of its kind.

The cargo-refrigeration plant is situate on the shelter deck, starboard side forward, just abaft the forward funnel hatch. There is an extensive range of cold chambers on the orlop decks forward. These have been insulated at the Clydebank Works with granulated cork in a similar manner to the provision rooms, already described. There are six chambers in

all, the largest ones being divided by longitudinal central bulkheads. They have all been fitted for the carriage of frozen meats and poultry, cheese, bacon, butter and fruits, and particularly for the carriage of chilled beef. The compartments are quite independent of each other, and can be supplied with brine for cooling at any temperature suitable to the cargo carried. The brine circulates through galvanized wrought-iron piping, and the chambers are fitted with meat rails and removable hooks for hanging chilled meat, and also removable side tables for the stowage of forequarters. As the machinery is at a considerable distance from the chambers, a brine distribution house has been fitted on the shelter deck near to the chambers, from which the regulation and distribution is controlled.

The machinery in this case also is of the carbonic-anhydride type, and special care has been taken to ensure silent running. The plant is in duplicate throughout, and is electrically driven. There are two horizontal gas compressors, each direct coupled to a powerful electro-motor. These motors have been specially designed and constructed for the purpose by Messrs. Booth-royd, Hyslop & Co., of Bootle, and are so arranged, by means of shunt regulation, that they can run at any desired speed from 40 to 110 revolutions per minute. The speed can be regulated with absolute ease by the turning of one hand-wheel only, the motor running at the same speed as the compressor, and no gear wheels whatever are used. The compressors—Webb and others' patents—embody several new features, which it will be of interest to mention. The outer casing, of soft cast steel, encloses and supports a liner of hard close-grained cast iron, which forms the working bore of the cylinder, but is easily withdrawable from either end of the casing. Two forged-steel headers, carrying the valves, are bolted, one to each end of the casing. The one at the front end is fitted with the stuffing box and gland, and that at the back end with the plug cover. The piston is fitted with metallic packing rings of special metal, very accurately turned and finished, and held in place by a patent split junk-ring head, which, while doing away with all

screws, keys and pins, absolutely secures the rings in place, so that they cannot get adrift as long as the piston is within the cylinder bore. The gland is also fitted with a particular form of metallic packing, and no leather cups are used. The valves and seats are of special hard steel; the valves lift vertically, are of large area, and have no springs whatever. The compressors are constructed so as to be capable of long continuous runs without stop; the absence of leather cups entirely does away with the necessity for frequent renewals of the packing.

The gas condensers are independent, are of the vertical type, and consist of soft lap-welded coils of wrought iron, galvanized on the outside, and contained in galvanized wrought-steel shells. The evaporators are similarly constructed to the condensers, ample facilities being provided for easy access to the coils for cleaning.

Two high-lift Gwynne centrifugal pumps circulate the brine. These are direct coupled to variable-speed electro-motors by the same makers as the main motors, so arranged that the speed of each pump can be regulated to suit the resistance to be overcome, this resistance varying somewhat, according to the number of chambers in use, and the quantity of brine being circulated. We have already mentioned a large variety of goods that may have to be carried in the chambers, some frozen and others chilled. Frozen goods, of course, require brine at a low temperature for circulation through the cooling pipes in the chambers, but when carrying chilled beef, for example, brine at a temperature suitable for frozen goods is altogether too cold, and, if circulated, would rapidly freeze the quarters, especially those stowed nearest to the pipes. The temperature must be regulated with great accuracy. The same remark is applicable to certain fruits and other chilled produce. A brine supply at an accurate and easily regulated temperature is, therefore, of great importance, and in the installation under review special means have been provided so that this can be secured. This warmer brine is circulated by an independent pump, and in the distribution room mentioned above special

duplex headers have been arranged so that either the coldest brine or the warmer brine can be supplied to the cooling pipes in any one or more chambers, according to the cargo carried, whether chilled or frozen.

Webb and others' patent brine attemperator consists of a simple three-ported slide valve, enclosed in a cast-iron casing, and attached to a screwed spindle with a hand-wheel, so that the movement of the valve over the ports can be accurately regulated as required. The valve is kept up to its face by a suitable coach spring. A branch from the cold-brine supply main is coupled to one end of the valve casing, and the return warmer-brine main is coupled to the other end, the mixed or attemperated brine escaping through the central port and pipe, which is connected to the warmer-brine pump. An overflow pipe is connected back to the evaporators.

In working the apparatus, the warmer-brine pump draws from the mixing or attemperating valve chamber, delivering to the headers already mentioned, thence through the pipes in the chilled chambers back to the attemperator, there to be mixed with any given proportion of the coldest brine necessary to lower its temperature to the required degree. A suitable and specially-constructed pyrometer indicates the temperature, both the valve handle and thermometer being carried outside the insulated brine room. The temperature can be regulated by means of the handle controlling the valve exactly as required, higher or lower to suit, and the control is positive. The brine circulation generally, as in the provision plant, is an entirely closed circuit. There are no brine tanks, except a small one for mixing brine in the first instance, for charging the machine or for adding a little from time to time. Any little air or foul gas in the system is automatically disposed of through a small vent pipe carried outside from each evaporator. Though there are a large number of independent circuits, no difficulty whatever is experienced in regulating each exactly as required. With the closed circuit there is no difficulty with air locks or aeration of the brine, and the system is surprisingly simple, clean and easy to work.

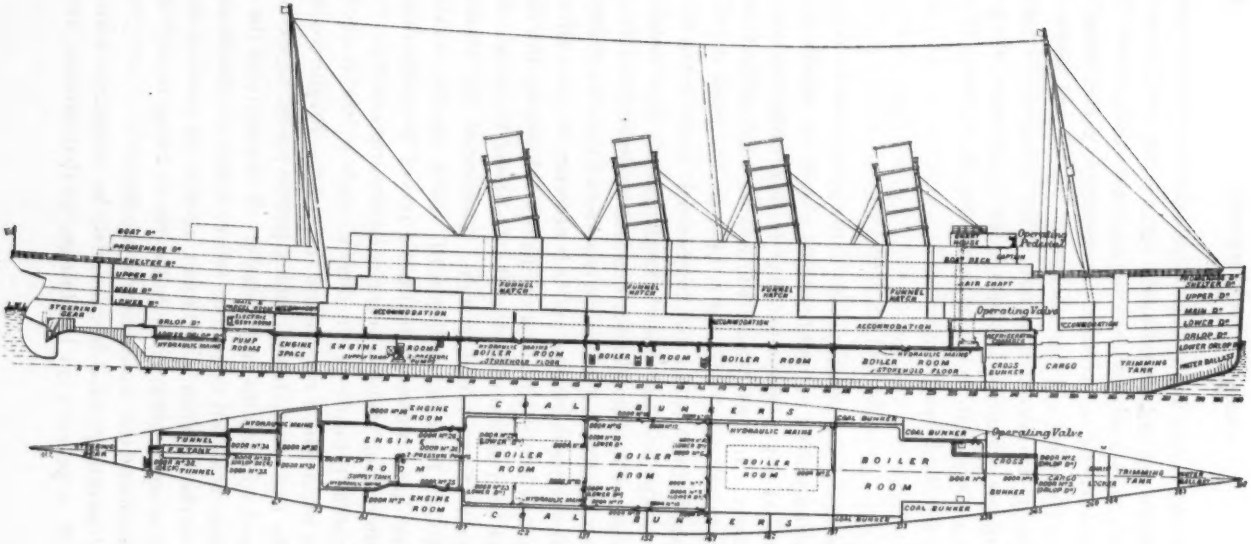
In both the cargo and provision installations the cold parts of the plants are not individually lagged, but are placed together in a well-insulated chamber or brine room, where they are always accessible, without the necessity of removing any lagging. The whole of the machinery, piping and fitting out of both plants has been carried out by the Liverpool Refrigeration Company, Limited, Liverpool, the installation being the latest of a large number of refrigerating plants fitted by this company for the Cunard Line.

SAFETY OF THE SHIP.

In describing the construction of the ship we have referred to the arrangement of bulkheads to ensure the safety of the vessel should any collision or other mishap cause the entry of the seawater. In this respect she will be much safer than almost any other vessel. It will be obvious, however, that if intercommunication is to be maintained freely between the various machinery and other compartments, these bulkheads must be pierced; otherwise serious and innumerable difficulties present themselves. Experience has proved that the absence of such doors is impracticable, and it is necessary to have effective, reliable, and above all, a quick method of closing the doors. This has been accomplished in the *Lusitania* by means of the well-known Stone-Lloyd system, constructed by Messrs. J. Stone & Co., Limited, of Deptford. After careful consideration they have preferred hydraulic power for actuating the doors, discarding steam, electricity, and compressed air.

Hydraulic pressure is supplied to each of the doors by a pressure main which runs round the vessel. Pressure is maintained by two steam-driven duplex pumps placed in the engine room, and continually under steam. A branch from the pressure main feeds an operating valve, which is placed on the casing of the forward boiler room, so that the pressure may be led into a small pilot main, called the "closing main," which also runs round the vessel to serve the doors.

The operating valve is connected by telegraph wires and chains to a pedestal on the bridge, so that pressure may be



ARRANGEMENT OF BULKHEADS AND WATERTIGHT DOORS,

led into the closing main, either from the operating valve itself, or by the pedestal on the bridge.

The doors are of the ordinary wedge type, and are formed of steel plate, suitably stiffened. They are each operated by an hydraulic cylinder, having two pistons connected by a rack, which gears with a pinion carried by a cross shaft; the shaft prolonged forms the door shaft, which, in turn, carries the driving pinion, gearing with the rack on the door. When space prohibits the fitting of the cylinder in the immediate neighborhood, intermediate shafting, with bevel wheels, can be arranged, and the cylinder can be placed in any convenient position.

The pistons are of slightly different sizes, so that a larger force is available to open the door, or bring it off its wedges. But the successful working of the system may be said to lie with the controlling valve, which is placed at each door. It consists of a tubular ram, which slides in a casing and is operated externally by a lever. The ram carries at its center an ordinary slide valve, which slides over three ports. These ports lead respectively to the opening and closing ends of the cylinder, and to an exhaust main, which runs round the vessel, and delivers into the supply tank. The pressure from the main is constantly behind the slide valve, so that, according to the position of the latter, the pressure flows either to the closing or to the opening end of the cylinder, the other end meanwhile exhausting. The ram is moved by a controlling handle, which is connected by a rod to the lever, and may be moved from either side of the bulkhead to which it is fitted. Thus the door may be opened or closed from either side of the bulkhead. The ram, at its lower end, runs through a U leather, and the closing main is connected to the space beneath it.

When the officer on the bridge moves over the pedestal handle, and thus opens the operating valve, pressure flows from the pressure main into the closing main, and thence to the under side of the ram, which is consequently forced over. The slide valve thus uncovers the closing port, and admits pressure to the closing end of the cylinder. To open the door

in such a case, it follows that the pressure on the ram must be relieved. This is accomplished in the following way: Inside the hollow ram is fitted a small miter valve, which is held on to its seat by the pressure in the closing main. A pilot spindle runs through the center of the ram, and terminates at one end against the miter valve, and at the other against the lever. Suitable packing against the spindle keeps it pressure tight. When the lever is moved towards the "open" position, it first depresses this spindle, which lifts the small miter valve off its seat and allows the pressure from the closing main to flow into the tubular ram, past the pilot spindle, which is fluted, and into the exhaust main by suitable ports in the ram and casing. The pressure on its lower end being now relieved, the ram can be operated, as before stated, to open the door. As soon as the handle or lever is released, the pressure in the closing main forces the miter valve back on to its seat, and moves the ram back to the "closed" position, and the door is again closed. Thus any man shut in a compartment may escape, and when he is through, the door will immediately close behind him. Warning of the closing of the doors is given by bells at each door ringing continuously as the door is moving.

In conjunction with the pumps and the pressure main is fitted a governor, on the spindle of which the pressure acts. Any increase over the working pressure of 700 pounds per square inch throttles the steam passing to the pumps. The pressure is thus kept to its normal height, and the pumps are ready at any moment to deliver up to their full capacity.

A circulating valve is also fitted in the pressure main, and allows a slight flow of water to pass into the exhaust main, and back to the tank. This device keeps the temperature of the water throughout the mains constant, thus preventing injury to the pipes, &c.

As it is desirable that the officer on the bridge should know the position of each door, whether open or closed, an electrical indicator is provided; this contains a fascia plate, on which a section of the vessel is engraved. Ruby discs are let into the

plate at different points, and are numbered to correspond with the doors they represent, and these are automatically lighted when the door is open.

THE PROPELLING MACHINERY.

In view of the immense power requisite to propel the new steamers at the designed speed, the Cunard Company, in September, 1903, called to their aid a committee composed of experienced and eminent engineers, to carefully consider the whole question of machinery design. After several months of deliberation and painstaking investigation this committee, in March, 1904, finally reported in favor of the adoption of turbine machinery of the Parsons type.

The steam-producing plant includes twenty-three double-ended and two single-ended boilers, arranged in four compartments. The main propelling machinery consists of two high-pressure ahead, two low-pressure ahead, and two astern turbines. The turbines are of the Parsons type, embodying the latest experience of the Hon. C. A. Parsons and the builders.

Owing to the immense size of these turbines, and in order to comply with the Admiralty's requirements as to subdivision, the main propelling and auxiliary machinery are located in nine different watertight compartments. In the

Table VIII.—LIST OF AUXILIARY ENGINES IN THE BOILER AND MACHINERY COMPARTMENTS.

In High and Low-Pressure Engine Rooms.

Feed pumps.....	Six pairs of Weir's..	18 in. and 13½ in. by 30 in.
Hotwell pumps.....	Four Weir's.....	12½ in. and 14½ in. by 30 in.
Surface heaters.	Two Weir's.....	1,750 sq. ft. each.
Contact heaters.....	Ditto	48 in. in dia. casing.
Feed-water filters.....	Two Harris's.....	36 in.
Ditto, auxiliary.....	Ditto	20 in.
Aux. circulating pumps..	Two Allen's 10-in...	36-in. disc engine, 7 in. by 6 in.
Auxiliary condensers.....	Two Clydebank.....	2,000 sq. ft. each.

Bilge pumps.....	Four Weir's.....	8 in. and 10 in by 21 in.
Fresh and condensed-water pumps	Two Carruther's.....	Two 6 in., two 6 in. by 6 in.
Water service circulating pumps	Ditto	Two 7½ in., two 10 in. by 12 in.
Sanitary pumps.....	Two Weir's.....	10 in. and 10 in. by 10 in.
Auxiliary air pumps.....	Ditto	10 in. and 22 in. by 12 in.
Oil pumps.....	Six Weir's.	7 in. and 8½ in. by 15 in.
Gland and jacket drain-tank pump	One Carruther's duplex	Two 4½ in. and two 5 in. by 5 in.
Sluice-valve engines.....	Two Clydebank.....	Two 6 in. by 6 in.
Reversing engines.....	Four Brown's.....	6 in by 8 in.
	In Pump Rooms.	
Circulating pumps.....	Two sets Allen's.....	Each set with two 18-in. by 10-in. engines and two 32-in. discharges, and two 42-in. discs.
Wet-air pumps.....	Four twin Weir's....	14 in. and 40 in. by 24 in.
Dry-air pumps.....	Ditto	7 in. and 24 in. by 7 in.
Aft water-service pump..	One Carruthers' duplex	Two 6 in., two 7 in. by 7 in.
	Evaporator Rooms.	
Wash-deck and fire pumps	One Weir's.....	10 in. and 10 in. by 10 in.
Evaporators.....	Six Liverpool Engineering Co.'s	
Distillers.....	Four Ditto.	

Pumps for evaporators: Two circulating pumps, two feed pumps and two brine pumps.

In Boiler Rooms.

Assistant feed and ash-ejector pumps	Four Weir's.....	14 in. and 10 in. by 14 in.
Ballast pumps.....	Two Carruthers' duplex	Two 8 in., two 10 in. by 10 in.
Ash ejectors.....	Eight sets Mechan.	
Ash hoists.....	Eight sets Crompton.	

Forced-draft fans and motors: Eight sets of fans and motors, each set with four fans and two motors; fans, 66 in. in diameter, Allen's.

Turbo-dynamo..	Four Parsons'.....	375 kilowatts.

largest and most central of these are located the two low-pressure and the two astern turbines, the whole of the feed pumps, hotwell pumps, oil pumps, and the pumps for the Stone-Lloyd hydraulic system of closing bulkhead doors. In each of the two wing compartments, separated from this central compartment by a longitudinal bulkhead, are placed respectively the high-pressure turbines, the auxiliary condensers, auxiliary circulating and air pumps, also the water-service, fire and bilge and fresh-water pumps. Aft of these, in separate wing compartments, are placed the evaporating and distilling plants, underneath which runs a portion of the high-pressure tunnel shafting. In the fifth compartment, extending right across the ship, are placed the four main condensers, and underneath these the feed tank. Aft of these again are two auxiliary-machinery rooms, each two-storied and each self-contained. In the lower are placed the main wet-air pumps, dry-air pumps and main centrifugal circulating pumps; and in each of the upper flats the electric generating stations, in which the power is supplied by two Parsons turbo-generators. Abaft these are the shaft tunnels.

The starting gear is situated on an upper platform at the forward end of the main engine room, and from this position practically everything can be controlled. The main regulating-valves for the high-pressure and the maneuvering turbines are arranged and worked very similarly to those on the *Carmania*, the Brown's engines being located on the platform, so that, in the event of any hitch, the valves can be worked at once by hand in place of steam power.

STEAM-GENERATING PLANT.

Although the turbines naturally awaken greater interest than the steam-generating plant, it will be more consistent with the natural order if, in entering upon a detailed description of the machinery, we deal first with the boilers.

There are, as already stated, twenty-three double-ended and two single-ended boilers in the ship, situated in four separate water-tight compartments. The forward boiler room, desig-

nated No. 1, has two single-ended and five double-ended boilers, and in each of Nos. 2, 3 and 4 boiler rooms there are located six double-ended boilers, in groups of three athwart the ship. In Nos. 2, 3 and 4 boiler rooms the bunkers are situated on the sides of the boilers and boiler casings; but in No. 1 boiler room, on account of the increasing fineness of the ship, recourse had also to be had to a large athwartship bunker, forward of which is a hold available either as a reserve bunker or for general cargo.

The double-ended boilers, are 17½ feet in diameter by 22 feet long. In each of these boilers there are 344 stay tubes and 720 plain tubes, the total being 1,064. The total heating surface is 6,593 square feet and the grate area 168.65 square feet. There are separate combustion chambers to each furnace, and the water and steam spaces are exceedingly ample. The single-ended boilers differ from the others only in their length.

The furnaces have a collective grate area of 4,048 square feet, while the total heating surface is 158,350 square feet. For each group of boilers there is a separate funnel, all being of the same dimensions, as the heating and grate surfaces of each boiler room are identical. These funnels are elliptical in form, the extreme dimensions of the outer funnel bing 19 feet by 26 feet, while the height above the grate level is 130 feet. We may add that the shells of the boilers are made of high-tensile steel, of a maximum tensile strength of 36 tons to the square inch; but the fronts and backs, as also all rivets, including those for the shells, are made of ordinary mild steel. The 192 furnaces are of the cambered type, which is a specialty of the Atlas Works, Sheffield, of Messrs. John Brown & Co., limited. These furnaces were made to a diameter over ribs of 4 feet and $1\frac{9}{16}$ inches, and for a working pressure of 195 pounds per square inch. The front and back ends of these furnaces were finished off in accordance with the latest practice, having their landings turned so as to ensure a perfect fit.

There are four main leads of steam piping, one from each boiler room and all four are carried right aft to the forward engine-room bulkhead, on the forward side of which a stop

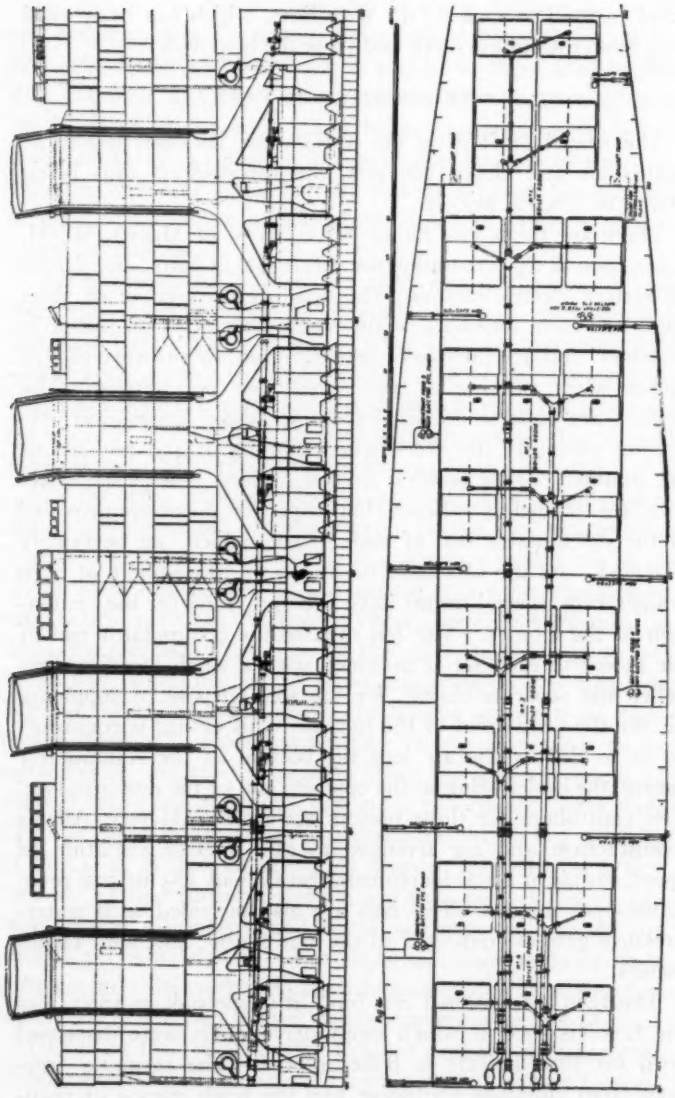
valve is fitted to each of the four lines, which can be worked both from the engine room and from the boat deck.

THE FORCED-DRAFT FANS.

The forced draft is on the well-known Howden system of heated air and closed ash pits, the forced-draft fans being driven by electric motors.

There are thirty-two motor-driven fans, of Messrs. W. H. Allen, Son & Co.'s manufacture, arranged in pairs, and driven by sixteen electric motors. The fans are throughout of sheet steel, and have impellers of the single-inlet type, 66 inches in diameter, and are capable of producing an air pressure of 3½ inches when running at 450 revolutions per minute. The motors are capable of developing 50 brake horsepower, receiving curent at 100 volts, when running at 450 revolutions per minute. These motors, also of Messrs. Allen's manufacture, are completely enclosed, but inspection doors are provided at the commutator end of each motor, which can be readily removed. As the sets have to run in an atmosphere at high temperature, special means have been provided for the ventilation of the motors. The fan next to the commutator end of the motor in each set is provided with a small auxiliary impeller and separate casing, for the sole purpose of supplying air for the ventilation of the motor. This casing is connected so as to deliver the air into the bottom of the commutator casing, the outlet being at the opposite end of the motor casing. The controllers for these motors are also of Messrs. Allen's manufacture, and are arranged to give a large variation of speed, rising in equal increments from about 185 to 500 revolutions per minute. The fans are also provided with water-pressure gauges connected to the fan casing, and with tachometers.

The twin-screw ship *Vera*, built at Clydebank in 1897, was the first instance in which steam-driven fans were dispensed with for forced draft in boiler rooms. The resulting freedom from vibration and noise, and the ready means of regulation from the starting platform, proved so satisfactory that



GENERAL ARRANGEMENT OF BOILERS OF THE "LUSITANIA."

motor-driven fans were subsequently installed, with equally good results, in the sister ship *Alberta,* built at Clydebank for the London and South-Western Railway Company, and the Steamship *Antrim,* built by John Brown & Company in 1905 for the Midland Railway Company. In these ships the controllers were fitted in the engine rooms and the fans entirely regulated from that position; but in the case of the *Lusitania* the distance from the engine room to the boiler rooms is so great that, while it was desired to be able to regulate the fans from the engine room, it was at the same time also considered advisable to retain the ordinary controllers attached direct to the fans in the boiler rooms, and this required result has been most satisfactorily attained by the very ingenious apparatus of Messrs. Siemens Brothers.

When it is desired to alter the position of any one of the fan controllers the operator works the switch handle of the sender to either "fast" or "slow," the handle, when released, being returned to the central position by a spring. This switch causes the motor on the controller to run either forward or backward, and through the gearing to turn the controller barrel either forward or backward. Geared to the controller barrel is the indicator switch, the indicator being situated just above the sender switch, and consisting of a revolving drum, on the periphery of which are figures and letters corresponding with those on the top of the controller case; this drum has an electro-magnetic control, worked from the indicator switch geared to the controller barrel, so that the figures or letters on the drum, as seen through the window of the indicator, will correspond with the position of the controller barrel. In order to ensure great accuracy of position of the controller barrel, a bell is placed in the indicator which is connected to a contact ring in the indicator switch, having portions of the ring cut away, so that the bell will continue to ring as long as the controller barrel is not in its exact position. It will thus be seen that the operator, when watching the figures on his indicator, can see exactly what is taking place on the controller, the bell telling him whether the controller is in its exact position or not.

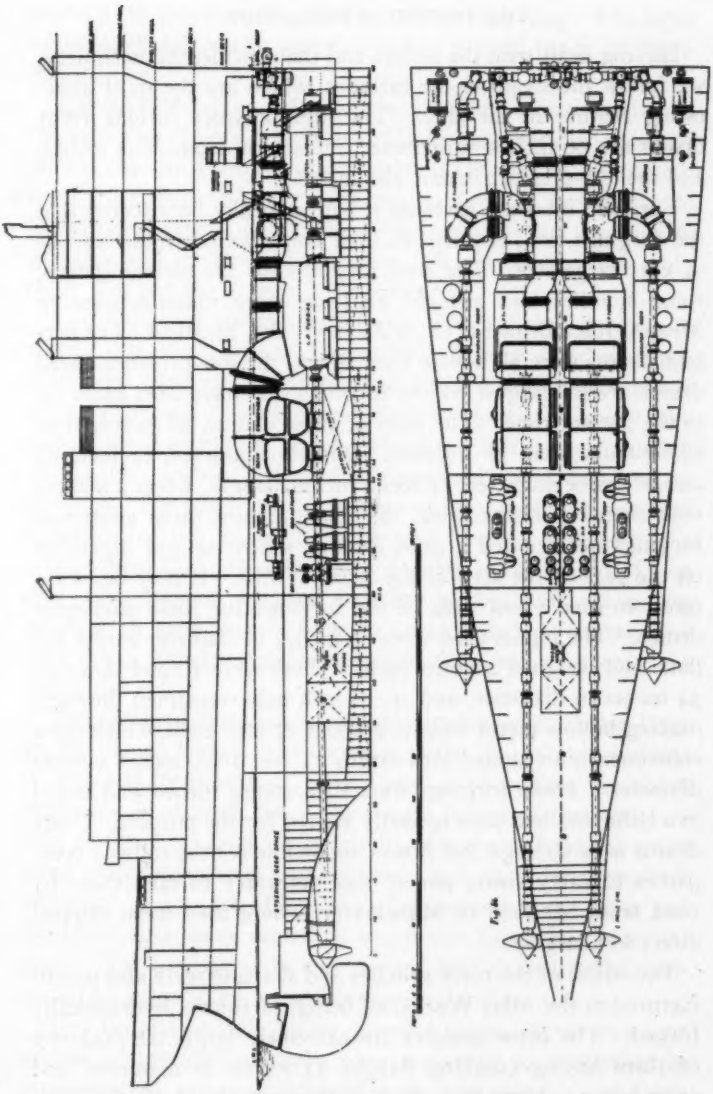
The working of the indicator is positive, not step by step, and the instrument is of the same type as is used in the Navy for controlling the firing operations. The motor on the controller is geared to the controller barrel by means of a slipping clutch, so that if the operator runs the motor beyond what is necessary to turn the barrel, no harm will be done. The whole apparatus is watertight. An ammeter is fitted above the indicator to record the amount of current used by the fan.

The air inlets to the fans have been arranged as trunks extending up to the boat deck, and, instead of the usual array of cowls, which are always unsightly, especially where, as in this ship, they require to be of great size, they are circular shafts fitted on top with hinged covers. The extent of the opening can be varied, and gear is also arranged so that the cover may be rotated to suit the wind when on the beam. When the vessel is steaming ahead the opening will be, as a rule, towards the bow.

THE DISPOSAL OF ASHES.

One of the difficulties which have to be dealt with in vessels burning such a large amount of coal as will be the case in this instance, is the disposal of the ashes. As a matter of fact, the expeditious disposal of the ashes has a direct effect on the steaming capabilities of any high-powered vessel, as it is impossible for the firemen to give proper attention to the firing when the stokeholds are hampered with ashes. Eight See's ash ejectors are fitted to the *Lusitania* (and also in the sister vessel), and with this apparatus all that is necessary is for the firemen to shovel the ashes into a hopper in the stokehold, after which they are dealt with by the apparatus without further manual labor on the part of the firemen, and are discharged twenty or more feet clear of the ship's side. Ten to fifteen minutes each watch will suffice to clear each stokehold of ashes.

An alternative system of dealing with the ashes, and for disposing of them when in harbor, has been fitted by Messrs. T. Albert Crompton & Co., London. This system is known as the Crompton atmospheric silent ash hoist.



GENERAL ARRANGEMENT OF TURBINES OF THE "LUSITANIA."

THE TURBINE INSTALLATION.

Having dealt with the boilers and their incidental machinery, we follow the steam to the turbines, which are the most interesting feature in the ship. The high-pressure turbine rotor drums are 96 inches in diameter, the astern drums 104 inches, and the low-pressure drums 140 inches.

For such immense turbines it will naturally be inferred that the forgings, etc., required in their construction would also be of very large size. The steel forgings for the turbine drums, rotor spindles and straight shafting were manufactured at Messrs. John Brown & Co.'s Atlas Works, Sheffield. The turbine drums were all hollow forged, and the low-pressure ahead drums are the largest hollow forgings that have been made up to the present time. The sizes of these drums, as delivered to Clydebank, were as follows: Outside diameter, 11 feet, 8⅜ inches; inside diameter, 11 feet 4 inches; length, 8 feet 2 inches; with metal 3 inches thick. To manufacture these enormous forgings necessitated a good deal of scheming and ingenuity on the part of the staff at the Atlas Works. It may be of interest to give a few data of the forgings for these particular drums. The ingots used were 60 inches in diameter across the flats, and weighed 42 tons each. These were forged down to 54 inches in diameter, and an 18-inch hole trepanned through, making hollow pieces weighing about 27 tons each, which were subsequently expanded into drums 11 feet 10½ inches outside diameter. These forgings were then rough turned and bored in a lathe that had been specially altered for the purpose. These drums were so large that it was impossible for the railway companies to carry them, and it was necessary to take them by road from Sheffield to Manchester, where they were shipped direct to Glasgow.

The whole of the rotor spindles and shafting were also manufactured at the Atlas Works, all being, of course, hydraulically forged. The rotor spindles are extremely large, the majority of them having coupling flanges 43 inches in diameter, and some being as large as 40 inches in diameter in the body. The intermediate shafting is 20 inches in diameter, and the tail

shafts 22⅛ inches in diameter by over 30 feet long. The rotor spindles, thrust shafts and tail shafts are all made of high-tensile steel.

The throttle valves, valve covers, strainers, expansion joints, etc., are also of very large proportions, and were made as steel castings by the Robert process, which is also one of the specialties of the Atlas Works. The uniformity and superiority of the physical results obtainable by this process are particularly striking, and are attained with less difficulty than by any other method at present in use.

The cast-steel dished wheels on which these drums are shrunk were supplied by Thomas Firth & Sons, Limited, Sheffield, who have made this particular type of turbine castings a speciality, and have supplied them to practically every firm of turbine builders in Great Britain. Each wheel weighed 11¾ tons. In all, Messrs. Firth supplied about 440 tons of castings. The whole of the castings were subjected to tests and inspection of the Board of Trade and Lloyds. Owing to the great contraction that takes place in steel—double that of cast iron—the greatest possible care has to be exercised in the molding of such huge wheels to avoid possible failures, and only long foundry experience and skill can overcome the difficulties.

The turbine blades vary from 2¼ inches to 22 inches long. In the longer blades the necessary radial and lateral stiffness is obtained by means of three rows of shrouding, in which expansion is allowed for. The longer blades of the low-pressure turbines are bound together by two circumferential strips laced with copper wire and soldered; and, in order to prevent distortion, due to the differences of expansion of the drum and the brass strip, very ingenious expansion joints had been devised. The binding strip was divided into short lengths, connected by brass tubes, in which they could slide.

The turbine spindle steam glands, valves, governors and system of lubrication which have answered so admirably in the *Carmania,* are again adopted in the new vessel.

The lifting-gear for the low-pressure turbine has to be capa-

ble of lifting the immense weight of 115 tons; and the provision of suitable apparatus and means of stowing the immense receiver and exhaust pipes was one of the many problems which have been so successfully solved by the builders of this vessel.

THE CONDENSERS.

The steam having performed its work in the turbine, the next step is to return it in the form of feed water to the boilers. In a leviathan such as the *Lusitania* the sequence of steps is practically similar to that in any ordinary installation, there being little scope of novelty in method. The number, size and arrangement of the auxiliaries form therefore the chief features which call for note.

There are four main condensers, arranged in pairs, each unit containing 20,700 square feet of cooling surface, giving an aggregate of 82,800 square feet. A 32-inch bore circulating-water pipe is led to each condenser from the large centrifugal pumps. The two auxiliary condensers, which are situated at the forward end of the high-pressure engine room, have a collective cooling surface of 4,000 square feet, and have separate circulating and air pumps.

Each of the four main condensers is fitted with the Harris-Anderson patent condenser-tube protector, supplied by the Harris Patent Feed-Water Filter, Limited, for preventing corrosion of the tubes, ferrules, etc. The principle of the system is the introduction into the circulating water of a metal which is electro-positive to the metal forming the tubes of the condenser, the tubes and the electro-positive metal being connected together. The tubes are saved at the expense of the metal that is connected to them. The composition of this protective metal may be altered to suit particular cases. The metal usually employed, however, is electro-positive to nearly all the various alloys of copper and zinc, and it, moreover, retains its protective properties till it is entirely dissolved. There is found to be no trouble with insoluble deposits on its surface. Though not always convenient, the apparatus is sometimes fitted inside the end of the condenser. The makers prefer, however, to

provide an independent vessel as a container for the protector, which vessel communicates with the water space by two pipes, which can be closed by valves when desired, thus enabling the protector to be inspected or renewed without interfering with the working of the condenser.

Each tube in the condensers of the *Lusitania* is brought into meallic contact with the tube plate by means of a soft-metal washer inserted on the top of the packing in the stuffing box. When the ferrule is screwed up the washer spreads out, thus forming the necessary contact between the tube and tube plates. Protective metal blocks are secured in the water ends of the condensers, some in direct contact with the tube plate, and others connected by cables to terminals at the opposite end to the blocks. Any corrosive action likely to attack the tubes is thus transferred to the protector, and pitting of the tubes, ferrules, etc., is avoided.

THE CIRCULATING PUMPS.

The circulating pumps, by Messrs. W. H. Allen, Son & Co., Ltd., Bedford, present some novel features. The main circulating engines consist of eight "Conqueror" type centrifugal pumps, having suction and discharge branches 22 inches in diameter, and arranged in four pairs, the discharge branches from each pair uniting into one common discharge of 32-inch diameter. Each pair of pumps is driven by a single-cylinder high-speed forced-lubrication engine of Messrs. Allen's well-known standard type, the engines again being arranged in pairs. Thus the main pumping machinery is grouped into two sets. The engine shafts can be coupled together in pairs, the engines running as two pairs of two-cylinder high-pressure engines, an arrangement of weights having been provided whereby the balance is exceedingly good under these conditions. The steam distribution is effected by means of piston valves, the valve chest being cast in each case in one piece with the cylinder, the whole being of exceedingly close-grained and tough metal. The cylinders have a diameter of 18 inches, with a stroke of 10

inches and together are capable of developing 350 brake horse-power at a speed of 300 revolutions per minute, receiving steam at 160 pounds per square inch, and working against a back pressure of 10 pounds per square inch. The cylinder bodies and covers are well lagged with silicate cotton, and neatly covered with burnished sheet brass, and fitted with the usual drain-cocks and relief valves, presenting a very smart appearance. Cast in one with each cylinder is a substantial cast-iron distance piece, which is faced square with the bore of the cylinder for bolting to the top of the engine trunk. This distance piece is provided with openings through which access can be obtained to the stuffing boxes, which are all packed with United States metallic packing.

As stated above, the cylinders are arranged in pairs, each pair standing upon a cast-iron trunk of very rigid design, which carries the slide faces for the crossheads: these faces are accurately scraped up and finished square with the top and bottom faces of the trunk. In front are three doors which can be readily removed for inspection and adjustment of the working parts. Special oil and water glands are fitted to the top of the trunk where the piston and valve rods pass through it. This effectually prevents the oil from working up to the cylinders from the crank chamber, and precludes water from the cylinders entering the crank chamber. The whole of the trunks and cylinders complete stand upon a rigid box-section bed plate, in which is arranged the oil reservoirs, filters and oil pumps, each of these latter fittings being arranged in duplicate, so that the engines may be disconnected from each other and run separately. The oil pump is of the valveless oscillating type, and is driven from the engine eccentric, and delivers oil under pressure to all the working parts. Each pump is also provided with an oil-pressure regulator, whereby the pressure can be regulated while running.

On the front of the engine trunk are arranged the oil and steam pressure gauges, in close proximity to the stop valves; and neat transmission gear is arranged for operating the drain cocks of the engine.

Each end of the bed plate is provided with an extension for

bolting to a similar extension of the gun-metal pump casing. An outer bearing is also provided between the flywheel of the engine and the pump. All the main bearings of the engine and pumps are of cast steel, lined with white metal, with the exception of the crosshead bearings, which are of gun metal. Separate barring gear is also provided for each flywheel. Owing to the engines being of the totally-enclosed type, tachometers are also provided for each set to continuously indicate the speed of the machinery.

As stated above, the main pumps are constructed throughout, both casings and impellers, of gun metal, the casings being 7-16 inch thick, and the discs having a diameter of 42 inches. The pump shaft is of forged bronze, and carried in bearings external to the pump. The shaft enters the pump casing at each end through stuffing boxes having gun-metal glands and special provision for lubrication.

In addition to the above main circulating pumps and engines two auxiliary circulating pumping engines have been fitted, each pump being of gun metal, and having suction and discharge branches 10 inches in diameter, while the diameter of the impeller is 36 inches. The engine is of Messrs. Allen's standard open type, having a single cylinder 7 inches in diameter, with a stroke of 6 inches. The steam distribution is effected by means of a piston valve. The cylinder, valve chest and cover are lagged with silicate cotton, and neatly covered with blue sheet steel. The engine-bed plate is rigidly connected to the casing of the pump, and external pump bearings are provided, being lined with white metal. The piston-rod and valve-rod glands are also fitted with metallic packing of the United States type, and lubrication is provided from a central oil box, from which oil is carried by pipes to the various bearings. The cylinder is fitted with the usual grease cup and spring relief valves and drains.

FEED-WATER PUMPS AND HEATERS.

To the condensers, four in number, are connected four Weir wet-air pumps, 40 inches in diameter by 24 inches stroke. These are of Messrs. G. and J. Weir's twin type, having two

steam cylinders, two pump barrels, with the pump rods cross connected by a beam. Steam is admitted to both cylinders by a single valve of the Weir pattern, designed specially for air-pump duty, but comprising the usual and distinguishing features of the well-known Weir valve. Gun metal has been adopted for the pump barrels, the buckets, foot and head valve seats, which latter are fitted with Kinghorn valves and gun-metal guards. The cylinders are supported on a cast-iron entablature set on angle wrought-iron columns. The piston rods are of steel, connected by a crosshead with the pump rods, which are of manganese-bronze, and work in vertical guides.

In addition to these wet-air pumps, which are capable of maintaining the requisite vacuum when the system is reasonably tight, provision is made for unexpected or accidental leakage by fitting four sets of Weir double dry-air pumps, 24 inches in diameter and 7 inches stroke, for dealing with air only. In these the air-pump chambers are situated over the steam cylinders of a double-connected enclosed high-speed engine. These chambers are of gun metal, and are of the single-acting type. The air passes into the barrel above the buckets through annular openings, and is forced through the head valves on the up-stroke of the pump. The compression of the air results in a certain rise of temperature, which is taken care of by a small supply of circulating water, which passes through the chamber and carries off the heat. Steam is admitted to the engine by a piston valve controlled by a governor fitted on the shaft in the usual manner.

From the air pumps the feed water passes to the hotwell, from which it is taken by four Weir hotwell pumps 14½ inches by 30 inches, of the firm's light-duty type, fitted with Kinghorn valves, and having gun-metal liners, brackets, and manganese-bronze rods. These pumps are automatically controlled by Weir control gear fitted in the hotwell, so that the speed of the pumps corresponds to the quantity of water passing into the chamber. The feed water is discharged by these pumps through two Weir surface feed heaters, where the ex-

haust steam from all the auxiliaries (with the exception of the turbo-generators) is utilized to heat the feed, and as this steam is impregnated with oil, it flows, after condensation, by gravity through an oil filter into the hotwell tank. In addition to this feed heater there are also fitted two Weir direct-contact heaters, into which the exhaust steam from the turbo-generators is led. There is here also control gear for regulating the speed of the main feed pumps. These consist of three pairs of Weir standard feed pumps, 13½ inches in diameter, with a 30-inch stroke, which are supplemented by a duplicate installation of auxiliary feed pumps of the same size and number. These pumps have all gun-metal barrels, with manganese-bronze valves and pump rods, steel piston rods, with the requisite suction and discharge stop valves for drawing from the feed heaters and discharging to the boilers.

In addition to these auxiliaries Messrs. G. & J. Weir, Limited, have also supplied four duplex pumps of special design for ash-ejector and auxiliary feed duty, 10 inches in diameter, with a 14-inch stroke, three duplex pumps for sanitary and wash-deck purposes, also four single direct-acting bilge-pumps, 10 inches in diameter, with a 21-inch stroke. For the supply of oil to the turbine bearings, six of their special direct-acting lubricating pumps are fitted. For dealing with the water and air from the auxiliary condensers, they have furnished two of their latest type of single direct-acting air pumps, known as the "Monotype" pattern, 22 inches in diameter, with a 12-inch stroke. These represent the latest developments in air-pump design. The installation of Weir auxiliaries is very complete and representative, and practically handles the feed water from the time it leaves the condenesr until it is returned to the boilers; a responsible duty which calls for most reliable equipment.

FEED-WATER FILTERS.

Two feed-water filters of the well-known Harris type, supplied by the Harris Patent Feed-Water Filter, Limited, are fitted in connection with the hotwell pumps of the main tur-

bines, and filter the water on its passage to the feed heaters. The filters are each 36 inches in diameter, and of gun metal throughout, the principal feature in their internal construction being the central sludge outlet—an ingenious arrangement, by which the filtering area is divided into eight separate sections, each of which can be sludged out independently of the rest, the whole force of the reversed current of the water when cleaning, being concentrated on only one-eighth of the surface, so that the cleaning is most efficient, and can be effected in a few minutes without the necessity of opening up. The filters present a most compact appearance, and everything is well arranged to facilitate their ready manipulation. Two smaller filters, 20 inches in internal diameter, also in gun metal, are fitted in connection with the auxiliary machinery. These filters are of the same type as the larger ones, but with all valves self-contained.

DISTILLERS AND EVAPORATORS FOR MAKING-UP THE FEED WATER.

The distilling machinery is of Quiggin's well-known type, and was manufactured by the Liverpool Engineering and Condenser Company, Limited, Brunswick Dock, Liverpool.

There are two complete sets of plant in the *Lusitania,* and these supply the whole of the distilled water required for all purposes, the total capacities of each plant being, for cooking and drinking purposes, 18,000 gallons per 24 hours; for baths and washing, 15,000 gallons per 24 hours; while the evaporators for feed make-up purposes for the boilers, when working compound effect, supply 240 tons per 24 hours, and when working single high-pressure effect, 350 tons per 24 hours.

Each plant consists of one evaporator for the production of distilled water, and two for feed-make-up purposes, the two latter being arranged to work in series—compound, or separately—single effect. All the evaporator shells are made of rolled naval brass, double riveted. This is for the purpose of reducing the weight as far as possible. The ends of the evaporators, all the mountings, as well as the frame and doors, are

constructed of gun metal. The coils are made so that they can be withdrawn bodily, and are all interchangeable, and the coils can be taken out separately for the purpose of cleaning and for inspection; while in order to facilitate this operation a complete spare heating surface is provided for each size of evaporator, in order that the coils may be replaced by a clean set, when required, in a few minutes. An automatic feed-water regulator is provided for each evaporator, and this maintains the water level in the evaporator at a constant height.

The condensers have coils of solid-drawn copper, and are tinned inside and outside; the coils can be withdrawn bodily with the cover by simply unscrewing a nut on the spigot end at the bottom connection to the filter. The sectional area of the coils diminishes from top to bottom, but each coil has a parallel surface throughout. The inlet for the steam is of full bore where the steam enters, but is gradually reduced in area to a crescent section, until at the outlet end it is only about one-third of the original sectional area. The volume of the steam is reduced as it condenses, and is kept in contact with the condensing surface, owing to the diminishing area of the coil. It is claimed that in this way the surface is rendered much more effective than it would be if the coils were of the same sectional area throughout. The filter, which is charged with animal charcoal and limestone chips, is in the base of the condenser. As a means of aerating the distilled water there is a pipe fitted, which is tapped from an iron-pipe connection, and there is a door for access to the filter. The circulating water enters and flows as shown. There are two condensers in each set. The shells of the condensers are of galvanized mild steel.

In each set of apparatus there are three pumps, namely, one vertical duplex circulating pump, one vertical duplex evaporator feed pump, and one vertical single direct-acting type brine pump for pumping the brine from the low-pressure evaporator (when working compound effect); after the water has been diluted and cooled with sea water it is pumped overboard. All these pumps are made with solid gun-metal water ends.

PUMPS FOR SUNDRY DUTIES.

In the engine room there are a great variety of pumps for sundry duties. Many of these have been supplied by Messrs. J. H. Carruthers & Co., Limited, Polmadie, Glasgow. The arrangement of framing in this type gives very free access to all the moving parts. All the important joints of the valve gear are adjustable. The water valves are easily examined through the front doors of the pump.

Among the pumps supplied are the following: Two for ballast service, with cylinders 8 inches and 10 inches by 10 inches; two for water service, with cylinders 7½ inches and 10 inches by 12 inches; two for washing decks 6 inches and 6 inches by 6 inches; one for sanitary service 6 inches and 7 inches by 7 inches. All of the pumps have gun-metal ends.

VENTILATION OF THE ENGINE ROOM.

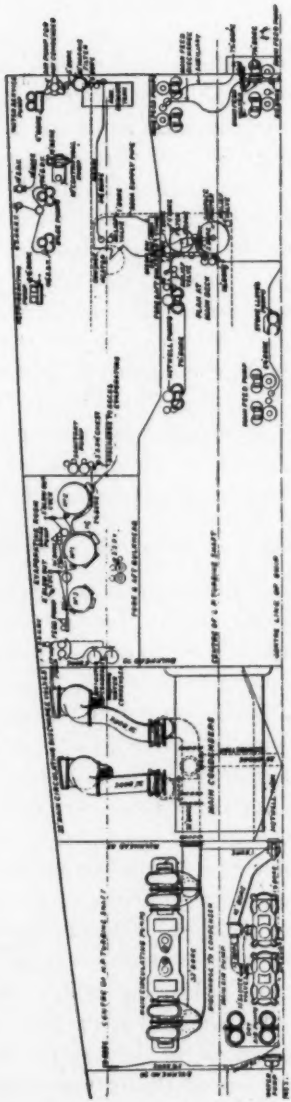
Messrs. Laurence, Scott & Co., Limited, Norwich, supplied twelve fans of 35 inches, two of 30 inches and two of 25 inches diameter, all electrically driven and adapted for the ventilation of the engine room. The outputs specified were respectively 26,000 and 14,000 cubic feet per minute, with free discharge at 315 and 450 revolutions per minute, the fans being direct driven and carried on an extension of the motor spindle. The company's standard type of semi-enclosed motor was adopted, fitted with gauze grids, the magnets being series wound for the reasons given below. In view of the high temperature of the situations in which some of these fans work, the motors were made large, and the temperature rise in a six hours' run was kept below 50 degrees F. The armature is all built up on a cast-iron quill, and is self-contained and independent of the shaft, on to which it is slipped when completed. Series winding was adopted for the magnets, as this gives better regulation of the load than shunt winding would do. The power required by a centrifugal fan at a constant speed goes up rapidly as the resistance to its free discharge is removed, reaching a maximum when disconnected altogether from its air trunks. The variation in speed of a series-wound

motor tends to correct the effect of variations in the resistance to discharge of the air, and keeps the load on the motor and the volume of air more nearly constant than would be the case if a shunt motor were used. The series winding also gives a simple method of speed control without the use of resistances. For slow speed all four field coils are arranged in series with each other and the armature. For full speed the field coils are arranged in two parallel circuits, each of two coils in series, these being still in series with the armature. The motor is then running with a lower resistance in series with the armature and with a weaker field, and therefore at a higher speed. The barrel controller is protected by an overload and no-voltage device. In the event either of an overload or failure of supply, the barrel carrying the contracts flies to the "off" position, even if the operating handle is being held "on." The fans are Messrs. Davidson & Co.'s make, of the well-known Sirocco type, and, like the motors, are amply large for the work.

ARRANGEMENT OF AUXILIARY MACHINERY.

The condensers are in a separate compartment abaft the main central engine room and over the two shafts driven by the low-pressure turbines. The circulating pumps and the air pumps are in a compartment abaft the condensers. The air pumps are accommodated between the shafts, and are at a lower level than the circulating pumps.

The evaporator and distilling plant are in a compartment in the wings of the ship, over the outer shafts and abaft the high-pressure turbine. In this way space is admirably economized. It will be understood by those who have studied the preceding figures that the high-pressure turbines in the wing compartments are not in the same athwartship line as the low-pressure turbines, the former being considerably in advance. This disposition of the turbines has enabled the larger of the auxiliary engines to be grouped in the forward part of the central engine compartment in the vacant space around the astern turbines, which are at the forward end of this center engine compartment. The main feed pumps occupy a central



GENERAL ARRANGEMENT OF AUXILIARY MACHINERY AND DISCHARGE PIPE.

position, and the hotwell pumps are in the wing; while to the forward of them, but on a higher level, are the filters and the feed-water heaters. Still further forward, against the main engine-room bulkhead, are the main feed pumps and valves.

SOME RECENT RECORD PERFORMANCES.

Record-breaking steamers.	Time.			Speed.	Best day's run.
	days.	*hrs.*	*min.*	*knots.*	
In 1840 *Britannia's* trip—Liverpool to New York	14	0	0	8½	...
In 1862 *Scotia's* trip—Liverpool to New York.	8	22	0	13	...
Servia, 1884.....Outwards	7	10	47	...	...
Homewards	6	23	57	...	...
Oregon, 1884.....Outwards	6	10	9	...	...
Homewards	6	16	59	...	...
America, 1884.....Homewards	6	14	18	...	...
Umbria, or *Etruria*.....Outwards	6	1	44	19.3	501
Homewards	6	3	12	19.1	...
Paris, or *New York*.....Outwards	5	14	24	20.7	530
Homewards	5	19	57	20.1	...
Campania, or *Lucania*, 1904..Outwards	5	7	23	21.82	562
Homewards	5	8	38	22.01	533
Kaiser Wilhelm der Grosse, 1902, Cherbourg-Sandy Hook	5	15	20	22.81	580
Kaiser Wilhelm der Grosse, 1901, Sandy Hook-Plymouth	5	10	0	23.0	553
Deutschland, 1903.....Cherbourg-New York	5	11	54	23.15	...
Deutschland, 1900.....New York-Plymouth	5	7	38	23.51	...
Kronprinz Wilhelm, 1902.....Cherbourg-Sandy Hook	5	11	57	23.09	581
Kronprinz Wilhelm, 1901.....Sandy Hook-Plymouth	5	8	18	23.47	561
Kaiser Wilhelm II, 1904.....Cherbourg-Sandy Hook	5	12	44	23.12	583
Kaiser Wilhelm II, 1906.....Sandy Hook-Plymouth	5	8	16	23.58	564

The *Deutschland's* westward mean speed of 23.51 knots, made over a long course, and not, therefore, a record in point of time, is equivalent to steaming from Queenstown to Sandy Hook in about 4 days 23 hours; and the *Kaiser Wilhelm II's* homeward mean speed of 23.58 knots would bring her to Queenstown in a few minutes' less time.

DIMENSIONS AND PERFORMANCES OF NOTABLE ATLANTIC STEAMERS.

Steamer's name.	Builders.	Date.	Molded dimensions.			Proportion of length.		Draught.	Displacement.	Gross tonnage.	Cylinders.		Boilers.			Indicated horsepower.	Speed on trial.
			Length.	Breadth.	Depth.	To beam.	To depth.				Diameter in inches.	Stroke.	Heating surface.	Grate area.	Working pressure.		
			ft.	*ft. in.*	*ft. in.*			*ft. in.*	*tons.*			*in.*	*sq. ft.*	*sq ft.*	*lb.*		*knots.*
Great Eastern	Scott Russell	1858	680	83 0	57 6	8.192	11.826	25 6	27,000	24,360	Screw, four 84-in.; paddle, four 74-in.	S., 48 P., 17	...	...	30	7,650	14.5
Britannic	Harland & Wolff	1874	455	45 0	36 0	10.111	12.640	23 6	8,500	5,004	Two 48-in., two 83-in.	60	...	...	70	5,500	16
Arizona	Fairfield Co.	1879	450	45 2	37 6	9.955	12.000	22 0	...	5,147	One 62-in., two 90-in.	66	...	...	90	6,300	17
Servia	Clydebank Works	1881	415	52 0	40 6	9.908	12.716	23 3	9,900	7,392	One 72-in., two 100-in.	78	27,483	1,014	90	10,300	17
Alaska	Fairfield Co.	1881	400	50 0	39 8	10.000	12.607	22 0	...	6,932	One 68-in., two 100-in.	72	...	...	100	10,500	18
City of Rome	Barrow Co.	1881	543	52 0	38 9	10.432	14.000	22 0	11,230	8,141	Three 46-in., three 86-in.	72	29,286	1,398	90	11,900	18.23
Aurania	Clydebank Works	1882	470	57 0	39 0	8.245	12.051	20 9	8,800	7,269	One 68-in., two 91 in.	72	23,284	1,021	90	8,500	17.5
Oregon	Fairfield Co.	1883	500	54 0	40 0	9.259	12.500	23 0	10,500	7,375	One 70-in., two 104-in.	72	38,047	1,428	110	13,300	18.3
America	Clydebank Works	1884	432	51 0	38 0	8.470	11 520	23 0	9,300	6,500	One 63-in., two 91 in.	66	22,750	882	95	7,354	17.8
Umbria	Fairfield Co.	1884	500	57 0	40 0	8 772	12.500	22 6	10,500	7,718	One 71-in., two 105-in.	72	38,817	1,606	110	14,321	20.18
Lahn	Fairfield Co.	1887	448	48 10	36 6	9.174	12.247	23 0	7,700	5,661	Two 32½ in., one 68-in., and two 85.in.	72	...	...	150	8,900	17.78
Paris‖	Clydebank Works	1888	528	63 0	41 10*	8.373	12.610	23 0	13,000	10,499	Two 45-in , two 71-in., and two 113.in.	60	50,265	1,293	150	20,100	21.8
Augusta Victoria	Vulcan Co , Stettin	1889	460	55 6	39 0	8.288	11.795	22 9	9,500	7,661	Two 41 5/16-in., two 66 1/16-in., and two 106 5/16-in.	63	36,000	1,120	150	14,110	18.31
Columbia	Laird Brothers	1889	463	55 6	39 0	8 333	11 860	22 9	9,500	7,578	Two 41-in., two 66-in., and two 101-in.	66	34,916	1,226	150	13,680	19.15
Teutonic	Harland & Wolff	1890	465	57 6	42 2	9.826	13.425	22 0	12,000	9,686	Two 43-in., two 68-in., and two 110-in.	60	40,072	1,154	180	19,500	21
Normannia	Fairfield Co.	1890	500	57 3	38 0	8.73	13.150	22 0	10,500	8,716	Two 40-in., two 67-in., and two 106-in.	66	46,490	1,452	160	16,352	20.78
Spree	Vulcan Co., Stettin	1890	463	51 6	37 6	9.00	12.346	22 0	8,900	6,963	Two 38-in., one 75-in., and two 100-in.	72	...	...	165	13,000	19.6
Furst Bismarck	Vulcan Co., Stettin	1891	503	57 3	38 0	8.777	13 224	22 6	10,200	8,000	Two 43½-in., two 67-in., and two 106½-in.	63	47,000	1,450	157	16,412	20.7
Campania	Fairfield Co.	1893	600	65 0	41 6	9.231	14 457	23 0	8,900	12,500	Four 37-in., two 79-in., and four 98-in.	60	82,000	2,630	165	30,000	22.01‡
St. Louis	Cramp, Philadelphia	1895	536	63 0	42 0	8.50	12.75	26 0	16,000	11,629	Four 28-in., two 55-in., two 77-in., four 77-in.	60	40,320	1,144	200	18,000	21.08‡
Kaiser Wilhelm der Grosse	Vulcan Co., Stettin	1897	625	66 0	43 0	9.46	14.27	28 0	20,880	14,349	Two 52-in., two 89-in., and four 96.4-in.	68.8	84,285	2,618	178	30,000	22.5 to 23†
Oceanic	Harland & Wolff	1899	685	68 5	49 0	10.01	13.98	32 6	28,500	17,274	Two 47.5-in., two 79-in., and four 93-in.	72	74,686	1,962	192	27,000	20.72
Deutschland	Vulcan Co., Stettin	1900	662.9	67 0	44 0	9.89	15.06	29 0	23,620	16,502	Four 36.61-in., two 73.6-in., two 103.9-in., and four 106.3-in	72.8	85,468	2,188	220	36,000	23.25 to 23.5‡
Kronprinz Wilhelm	Vulcan Co., Stettin	1901	663†	66 0	43 0	...	...	29 0	21,300	15,000	Four 34.2-in., two 68.8-in., two 98.4-in., and four 102.3-in.	70.8	93,685	2,702	213	36,000	23 25 to 23.5‡
Kaiser Wilhelm II	Vulcan Co., Stettin	1903	678	72 0	52 6	9.41	12.9	29 0	26,000	20,000	Four 37.4-in., four 49.2-in., four 74.8-in., and four 112.2-in.	70.86	107,643	3,121	225	38,000	23 5‡
La Provence	St. Nazaire Works	1906	597 ft. 1½ in.	64 7½	41 8	9 23	14 35	26 9	19,160	13,750	Two 47.2-in., two 76.2.in., and four 88.18-in.	66.9	58,342	1,571	200	30,000	22.05
Lusitania	Clydebank Works	1907	769	88 0	60 4½	8.65	12.56	33 6	38,000	32,500	Turbines.	...	158,350	4,048	195	68,000	25

COAL CONSUMPTION OF ATLANTIC CUNARD LINERS.

.....	*Britannia*, 1840.	*Persia*, 1856.	*Gallia*, 1879.	*Umbria*, 1884.	*Campania*, 1893.	*Lusitania*, 1907.
Coal necessary to steam to New York, tons.....	570	1,400	836	1,900	2,900	5,000*
Cargo carried, tons.....	224	750	1,700	1,000	1,620	1,500
Passengers.....	115	250	320	1,225	1,700	2,198
Indicated horsepower.....	710	3,600	5,000	14,500	30,000	68,000
Steam pressure, pounds..	9	33	75	110	165	200
Coal per indicated horse-power per hour, pounds.	5.1	3.8	1.9	1.9	1.6	1.45*
Speed, knots.....	8.5	13.1	15½	19.0	22.0	25.0

* Estimated.

—"Engineering."

INERTIA AND TORSIONAL STRESSES AND PRESSURES ON BEARINGS, TOGETHER WITH AN INVESTIGATION OF THE LUBRICATION PROBLEM, OF THE PORT MAIN ENGINE U. S. S. *TENNESSEE.*

By Lewis Hobart Kenney, B. S., M. E. Associate.

The following computations and investigations were undertaken as a means of determining the causes of the heating of the crank-pin and main bearings, on the supposition that perhaps the pressures on the bearings were excessive. The crank pins have given the most trouble, while the main bearings have given but little. It is assumed that the engines were correctly lined up, and no allowance for friction of the engines has been made in the computations. The pressures on the crank pin are the resultant of the steam and inertia forces. One-half the connecting-rod weight has been considered concentrated at the crank pin, as a revolving weight, and its vertical component, corrected for weight due to gravity, has been added to the inertia force of the reciprocating weights for the crank-pin pressures. The revolving weight of the crank pin and crank webs has been considered in the computation of main bearing pressures.

The inertia forces of the valve gear were computed and the effect on the main bearings considered. This inertia force was reduced to an equivalent tangential force at crank radius, and a diagram was drawn to show the combined or resultant tangential force. The area included between this curve and its axis shows a positive area of 0.05 square inch. Of course, this should equal zero and the error may be largely graphical. This area reduced to a force at crank radius is

equal to 16.7 pounds which, compared to the mean tangential force of the engine, 268 000 pounds, makes the percentage error very small. All the computations were made by the writer, with the co-operation of Mr. William B. Robins.

The computations for bearing pressures give for the maximum crank-pin pressures 570 pounds per square inch of projected bearing surface, which is not excessive with efficient lubrication. Hence, after completing the computations the next subject to investigate was the system of lubrication.

The following analyses were made of the normal bearing metal used on the U. S. S. *Tennessee* and of "flakes" taken from the bearings. These flakes were found on the sides of the crank-pin brasses, near the distance pieces, and had apparently been drawn from the crown of the top and bottom brasses. Melting points and specific heats of the several elements are given, and, it seems, that the percentage of the elements of the flakes drawn from the normal metal bear a close relationship to their specific heats. The explanation suggested is that the bearing was not properly lubricated and, for a short time at maximum pressure, there was metallic contact. This condition, combined with the high surface velocity of the crank-pin surface (667 feet at 127 R.p.m.) generates heat and will raise the temperature of the bearing metal. The continued application of heat will melt the surface of the bearing metal and gradually draw it out into the form of flakes, which are deposited at the place of no bearing pressure—that is, near the distance pieces of the brasses.

ANALYSES OF ALLOYS *TENNESSEE* PORT H.P. CRANK PIN.

	Copper.	Tin.	Antimony.	Lead.
Specification	3.70	88.8	7.50	.0
Normal metal	3.70	89.57	6.66	0.07
Flake	1.12	94.80	3.95	0.14
Fused piece, $\frac{1}{64}$ inch thick	1.00	95.37	3.57	0.03
Melting point (F)	1929.	442.	810.	618.
Specific heat	0.0951	.0562	.0508	.0314

The oil film between the crank pin and its brass must sustain the full pressure of the piston load, otherwise we get metallic contact; and, if there is an opening in the brass at

this point of maximum film pressure, the film will relieve itself and flow out of the oil hole. This is illustrated by the oil flowing back in the oil tubes on the after face of the thrust shoes and flowing down the tube leading to the forward face of the thrust shoe. The oil films sustain the thrust in keeping the collar and shoe apart, and the oil hole provides a place for the film pressure to relieve itself; and the oil easily flows to the place of no bearing pressure. In the case of the crank pin, the oil is led to the top of the crank-pin brass and the point of film pressure is on the center line of the connecting rod; hence, from the above theory it would seem that this is not the best place for the oil holes. The centrifugal oiling device is feeding oil against the crank-pin oil film pressure during the upper part and the lower part of the stroke, which are points of maximum film pressure, and it would seem that this is not an especially efficient method. The pressure of the brasses alternates, which allows, to some extent, for admission of the lubricant. These oil tubes, as installed on the connecting rods, are open at the top for the drop feed, and the oil in these tubes would be subject to same inertia forces as the connecting rod itself. The inertia force is greater at the head end than at the crank end of the stroke and, this combined with the rapid alternations when running at high speed, would interfere with the gravity flow of the oil. There might be used a telescopic oiling device as fitted for the crosshead pin, to lead the oil to the crank pin. This would give a "head" from the top of the cylinder to the crank pin, which would overcome the inertia effect of the oil in the connecting-rod oil tubes and insure a positive flow to the crank pin, the oil for the crank pin to be led, instead of to the top of the brass as at present installed, to an oil reservoir formed by the distance pieces at the sides of the crank-pin brass, care being taken to fit the distance pieces so that they will be reasonably oil tight. This is the point of no bearing pressure; hence, no film pressure. The telescopic oil tube for this might be concentric with the crosshead-pin tube, and the casting on the crosshead pin would distribute the oil from

this center tube to the crank pin. This would make the device as compact as the present oiling device for the crosshead pin. The oil holes may be drilled diagonally through the upper brass from where the present oil pipes enter it, to these reservoirs. The holes in the top of the brass now used would be filled to agree with the theory that the oil film will relieve itself by flowing out of the holes. The oil will not flow against this film pressure unless it had a gravity "head," or was fed in under pressure in excess of the film pressure; this would apply to the down stroke, of course. The operation of this installation would be as follows: As the crank pin goes down, the oil in the inboard reservoir will be thrown against the crank pin by centrifugal force and adhesion to the crank pin will carry the oil over to the place of bearing pressure in the upper brass. On the up stroke, the oil in the other reservoir would be thrown against the crank pin by centrifugal force and adhesion will carry the oil around to lubricate the bottom brass, which is carrying the pressure.

In many cases an oil which saponifies is used. This saponified oil might accumulate in these distance-piece reservoirs and obstruct the supply of oil to the crank-pin. Of course, a mineral oil without any saponifying content might be used.

The several quantities computed are as follows:

Cylinders:

1. Resultant vertical component of the revolving weight.
2. Resultant vertical inertia force of reciprocating weight.
3. Resultant combined vertical force of 1 and 2.
4. No. 3 reduced to equivalent pressure per square inch of piston area.
5. No. 4 plus steam pressure reduced to total resultant effective pressure.
6. No. 5 reduced to connecting-rod force.
7. No. 6 reduced to pressure per square inch of crank-pin brass projected bearing surface.
8. No. 6 reduced to tangential force at crank radius.
9. No. 6 reduced to normal force at crank radius.
10. No. 5 reduced to crosshead pressure, total and per square inch of bearing surface.

Valve Gear:

1. Resultant vertical component of revolving weight.
2. Resultant vertical inertia force of reciprocating weight.
3. Resultant combined vertical force of 1 and 2.
4. No. 3 reduced to a tangential force at eccentric radius.
5. No. 4 reduced to equivalent force at crank radius.
6. No. 3 reduced to pressure per square inch of projected area of eccentric bearing surface.
7. For backing gear No. 1, 2, 3, 4, 5 and 6 were computed.

Main Bearings:

1. One-half the vertical effective force delivered by the piston.
2. Valve-gear resultant vertical force.
3. Vertical component of the centrifugal force of the crank pin and crank webs corrected for weight due to gravity.
4. Algebraic sum of 1, 2 and 3.
5. Algebraic sum of 1 for two cylinders if the bearing is between them.
6. 4 and 5 reduced to pressure per square inch of projected bearing surface.

The following symbols were used in the computations:

Cylinder.	Valve gear.		
A		Crank angle in degrees (0° at top of stroke).	
	x	Eccentric angle in degrees (0° at top of stroke).	
B		Connecting-rod angle in degrees.	
	y	Eccentric-rod angle in degrees.	
L		Length of connecting rod in feet	$= 8.0$
	M	Length of eccentric rod in feet	$= 8.25$
r		Radius of crank in feet	$= 2.0$
	m	Radius of eccentric in feet	$= 0.437$
n		Ratio, $\frac{L}{r} = 4$	
	E	Ratio, $\frac{M}{m} = 18.86$	

Cylinder.	Valve gear.	
F		Centrifugal force of one-half connecting-rod weight in pounds.
	v	Centrifugal force of eccentric and strap in pounds.
Fv		Vertical component of F.
	V	Vertical component of v.
G		Weight due to gravity of one-half connecting rod in pounds.
	l	Weight due to gravity of eccentric and strap in pounds.
F_c		Resultant vertical component of $F = G \pm F_v$
	V_v	Resultant vertical component of $v = l \pm V$
g		Weight due to gravity of reciprocating parts in pounds.
	H	Weight due to gravity of reciprocating parts in pounds.
f		Inertia force of reciprocating weights in pounds.
	K	Inertia force of reciprocating weights in pounds.
f_v		Vertical component of f.
	k	Vertical component of K.
P_R		Resultant vertical force of reciprocating weights, in pounds $= g \pm f_v$
	M	Resultant vertical force of reciprocating weights $= H \pm k$
P_F		Resultant vertical inertia force of all weights $= F_c + P_R$
	p	Resultant vertical inertia force of all weights $= M + V_v$
	t	Tangential force at eccentric radius equivalent to p.
	Q	Equivalent force of t at crank radius.
A_p		Area of top of piston in square inches.
a		Area of bottom of piston in square inches.
P_{FA}		P_F referred to piston area as equivalent pressure in pounds per square inch.
P_{Fa}		Effective pressure delivered by piston, in pounds per square inch.

Cylinder.	Valve gear.	
P		Effective pressure delivered by piston in pounds.
	E_p	Pressure per square inch of projected bearing area on eccentric due to p.
	h	Weight due to gravity of backing eccentric rod and one-half of link.
	q	Inertia force of backing gear.
	U	Vertical inertia force of backing gear.
	o	Resultant vertical force corrected for gravity of backing gear.
	O	Resultant vertical component of backing gear.
	t_e	Tangential force at eccentric radius equivalent to O.
	Q_e	Equivalent force of t_e at crank radius.
R		Pressure on connecting rod due to P.
T		Tangential pressure at crank pin due to R.
N		Normal pressure at crank pin due to R.
C		Crosshead pressure due to P.
C_a		Crosshead pressure per square inch of bearing surface.
K		Crank-pin brass pressure per square inch of projected area due to R.
P_d		Piston displacement from top of stroke in decimal of stroke.
W		A weight in pounds.
M		A mass corresponding to W.
D		Revolutions per minute.
s		Vertical component of centrifugal force of crank pin and crank web in pounds.
S		Vertical component of centrifugal force of crank pin and crank webs, in pounds, corrected for gravity.

COMPUTATIONS.

Derive formula for computing angle B for any angle A; refer to diagram on plot.

(1) $$\frac{L}{r} = n = 4 \text{ for this engine.}$$

$$DE = r \sin A = L \sin B$$
$$r \sin A = nr \sin B$$

(2) $$\sin B = \frac{\sin A}{n} = \frac{\sin A}{4}$$

Derive formulae for computing connecting rod, tangential, normal, and crosshead forces due to piston force P.

(3) $$R = \frac{P}{\cos B} = P \sec B$$

$$T = R \cos RDT$$
$$\text{angle } RDT = 180 - (A + B) - 90$$
$$= 90 - (A + B)$$
$$\cos (90 - (A + B) = \sin (A + B)$$

(4) $$T = R \sin (A + B)$$

(5) $$= P\frac{\sin (A + B)}{\cos B}.$$

$$N = R \sin RDT$$

(6) $$= R \cos (A + B)$$

(7) $$= P\frac{\cos (A + B)}{\cos B}.$$

(8) $$C = P \tan B.$$

Derive piston displacement and inertia formulae.

Piston displacement is computed from the top of the stroke and for this position angles A and B are zero.

$$P_d = Ew + Ez$$
$$= r(1 - \cos A) + L(1 - \cos B)$$
(9) $$= r(1 + n - \cos A - n \cos B)$$

Subt. $n = 4$

(10) $$P_d = r(5 - (\cos A + 4 \cos B)).$$

In equation (9) substitute the value of $\cos B$ in terms of angle A.

From (2) $sin\ B = \dfrac{sin\ A}{n}$

$$sin\ B = (1 - cos^2 B)^{\frac{1}{2}}$$

$$(11) \qquad cos\ B = \frac{1}{n}(n^2 - sin^2 A)^{\frac{1}{2}}$$

Subst. Equa. (11) in 9

$$P_d = r\,(1 + n - cos\ A - (n^2 - sin^2 A)^{\frac{1}{2}})$$

$$\text{(Velocity)}\ \frac{d\,P_d}{dA} = r\left(sin\ A + \frac{sinA\ cosA}{(n^2 - sin^2 A)^{\frac{1}{2}}}\right)$$

(12) (Acceleration)

$$\frac{d^2 P_d}{dA^2} = r\left(cos\ A + \frac{cos^2 A - sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}} + \frac{sin^2 A\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right)$$

$$(13) \qquad = r\left(cos\ A + \frac{n^2\ cos^2\ A}{(n^2 - sin^2 A)^{\frac{3}{2}}} - \frac{sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}}\right)$$

$$(14) \qquad \text{Let } Z_1 = \frac{n^2\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}} - \frac{sin^2 A}{(n^2 - sin^2 A)^{\frac{1}{2}}}$$

$$(15) \qquad \text{then, } \frac{d^2 P_d}{d\,A^2} = r\,(cos\ A + Z_1).$$

Equation (12) will reduce to

$$\frac{d^2 P_d}{d\,A^2} = r\left(cos\ A + \frac{n^2\ cos^2 A - sin^2 A\ (n^2 - sin^2 A)}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right)$$

$$(16) \qquad \text{Let } Z_2 = \frac{n^2\ cos^2 A - sin^2 A\ (n^2 - sin^2 A)}{(n^2 - sin^2 A)^{\frac{3}{2}}}.$$

The value Z_2 may be a little more convenient for computation than Z_1.

Equation (12) will also reduce to

$$\frac{d^2 P_d}{d\,A^2} = r\left(cos\ A + \frac{cos\ 2\ A}{(n^2 - sin^2 A)^{\frac{1}{2}}} + \frac{sin^2 A\ cos^2 A}{(n^2 - sin^2 A)^{\frac{3}{2}}}\right).$$

If n is very large, as in the case of eccentric rods, $sin\ A$ will never be very large so that the last term may be omitted, and the second term will reduce to $\dfrac{cos\ 2A}{n}$. The greater the

value of n the nearer this approximate value will be to the value of Z_1 or Z_2 and (12) reduces to

$$\frac{d^2 P_d}{d A^2} = r\left(\cos A + \frac{\cos 2 A}{n}\right). \tag{17}$$

The angular velocity $= \frac{d A}{d t} = \frac{2 \pi D}{60}$

$$\frac{d A^2}{d t^2} = \left(\frac{\pi D}{30}\right)^2 \tag{18}$$

Subst. (18) in (15).

$$\frac{d^2 P_d}{d t^2} = \left(\frac{\pi D}{30}\right)^2 r (\cos A + Z_1). \tag{19}$$

We have the general formula: force = mass × acceleration.

Substituting (19) in this formula

$$\text{Force} = M\left(\frac{\pi D}{30}\right)^2 r (\cos A + Z_1)$$

Substituting $\frac{W}{g} = M$;

$$\text{Force} = \frac{\pi^2}{900 g} W r D^2 (\cos A + Z_1)$$

$$= 0.000\,341\ W r D^2 (\cos A + Z_1) \tag{20}$$

The total weight for the reciprocating force is the sum of the weights—piston, piston rod, crosshead, and one-half the connecting rod. The other half of the connecting rod is considered concentrated at the crank pin as a revolving weight.

The inertia of the reciprocating weights is computed by (20) and corrected for the weight due to gravity; the centrifugal force of the revolving weight is computed and its vertical component is corrected for the weight due to gravity. The algebraic sum of these two quantities is the total resultant inertia force.

The computations are all mathematical and only the results graphical, in order to insure accuracy and speed. By

dividing the connecting-rod weight into one-half as a reciprocating weight and the other as a revolving weight concentrated at the crank pin makes a convenient method of disposing of this troublesome problem. This method is used by Naval Constructor D. W. Taylor, U. S. N., in Volume 9, Transactions Society of Naval Architects and Marine Engineers; Sanford A. Moss' paper, "Influence of Connecting Rod upon Engine Forces," Volume 26, Transactions American Society of Mechanical Engineers, and Dr. D. S. Jacobus' paper, "Counter Weights for Large Engines," Volume 26 of the same society.

The mathematics for this investigation are developed from Dr. D. S. Jacobus' paper which appeared in Volume 11, Transactions of the American Society of Mechanical Engineers.

CYLINDER WEIGHTS, IN POUNDS.

Reciprocating.	H.P.	I.P.	L.P.
Piston	1 852	3 485	4 300
Piston rod	988	988	833
Crosshead	3 112	3 112	2 246
½ Connecting rod	3 600	3 600	2 940
Total	9 552	11 185	10 319
For computations (Total)	9 550	11 190	10 320
Revolving weight	3 610	3 610	2 940

VALVE GEAR.

The weight of the valve gear was divided into a reciprocating and revolving weight. The revolving weight was the eccentric and eccentric straps; the eccentric rod, one-half the suspension link, and all the reciprocating parts were taken as reciprocating weights for the ahead valve gear. For the backing gear the revolving weight was the same as above and the eccentric rod and other half the suspension link as the reciprocating weight. There may be an error in this assumption but perhaps it is sufficiently approximate for the purpose.

WEIGHT, IN POUNDS OF VALVE GEAR.

	Total.	AHEAD. Reciprocating.	AHEAD. Revolving.	ASTERN. Reciprocating.	ASTERN. Revolving.
H.P. CYLINDER					
1 Eccentric	700	...	700	...	700
1 Eccentric strap	740	...	740	...	740
1 Eccentric rod	375	375	...	375	...
1 Piston valve	1 155	1 155	...	...	...
1 Valve stem and balance piston	220	220	...	...	...
1 Valve-stem crosshead	200	200	...	...	...
1 Link	550	275	...	275	...
Total	3 940	2 225	1 440	650	1 440
Upward force of balance piston	...	1 902	...	...	...
Unbalanced force	...	323	...	...	...
I.P. CYLINDER	5 480	3 765	1 440	650	1 440
Upward force of balance piston	...	3 804	...	...	...
Excess unbalanced force	...	39	...	...	...
L.P. CYLINDER	5 920	4 205	1 440	650	1 440
Upward force of balance piston	...	5 474	...	...	...
Excess unbalanced force	...	1 269	...	...	...

CURVES PLOTTED.

The first curves drawn are for the piston displacement. The H.P. is drawn from the top of its stroke through one revolution, and the other cylinders are laid down in proper crank phase. The ratios of the displacements are equal to the cylinder volume ratios; the clearances and receiver volumes are ratios of H.P. cylinder volume. Valve displacement curves laid down in proper phase, together with the position of the port, show the position of cut off, release, compression and admission. The corresponding piston displacement may be scaled from the piston displacement curve. The volume the steam occupies may be scaled from the piston displacement curves; adding the clearance and receiver volumes will give the total volume of steam.

The valve displacement curve is plotted from equation (9) computed from top of stroke when x and y = zero.

$$P_d = r(1 + n - \cos A - n \cos B)$$

Substituting in (9),

Valve displacement $= m\,[19.86 - (\cos x + 18.86 \cos y)]$

This curve is compared to a sine curve as also is the piston

displacement, in the computed data, to determine the error if a sine curve were used.

After the piston and valve displacement curves had been laid down the actual indicator cards were plotted to correspond to the crank angles. Beginning at the top of the stroke the steam line of the top card is drawn for 180° and from there on the back pressure line of the card, in the same way the bottom card is drawn. The difference between these two lines is the effective pressure.

REDUCTION OF INDICATOR CARD PRESSURE TO EQUIVALENT PRESSURE PER SQUARE INCH OF PISTON.

Symbols:

Top P = Absolute pressure from indicator card.

A = Area of piston in square inches.

P_1 = Equivalent effective pressure per square inch of piston.

Bottom p = Absolute pressure from indicator card corresponding to P.

a = Area of piston in square inches.

p_1 = Equivalent effective pressure per square inch of piston.

K = Effective load on piston.

r = Ratio $\frac{A}{a}$

$PA - pa = K$

$(Pr - p)a = K$

$$P_1 = \frac{K}{A}; \qquad p_1 = \frac{K}{a}$$

$$= \frac{(Pr - p)a}{A} \qquad = \frac{(Pr - p)a}{a}$$

$$= P - \frac{p}{r} \qquad = Pr - p$$

H.P.	$P_1 = P - .950p$	$p_1 = 1.052P - p$
I.P.	$= P - .985p$	$= 1.017P - p$
L.P.	$= P - .987p$	$= 1.013P - p$

This method will give the correct effective pressure, but the curves as drawn give only the actual difference between the steam pressure of one card and the corresponding back pressure of the other. The result is that the top pressures are a little less and the bottom pressures are a little more, than the correct effective pressure. The mean of the H.P. error for the top and bottom is 5.5 pounds and the error of the other cylinders will of course be less.

REDUCTION OF INERTIA PRESSURE TO AN EQUIVALENT PRESSURE PER SQUARE INCH OF PISTON.

The total inertia force was first computed and the force exerted upward was divided by the area of the top of the piston. This will give the pressure required to cushion the inertia force and the pressure required to accelerate the piston on its return stroke. The pressure on the bottom of the piston was computed by the same method. The inertia curve laid down from this data will be fair, with no change at ends of stroke, and it is necessary to compute values for only 180° instead of 360°. The method involves a slight error because, starting from the top of the stroke, the inertia force is negative and about midstroke changes to a positive force, hence the inertia force on the down stroke should be divided by the top area and the bottom pressure determined by the same method. This pressure added (algebraically) to the indicator-card pressure will give the total effective pressure acting on the top or bottom of the piston. The inertia curves are correct from the beginning of the stroke to the point where the inertia force changes sign, and from there to the end of the stroke are in error to the extent of the ratio of the top and bottom areas of the piston. The error in determining the effective steam pressure by the method used is partly compensated for by the error in the inertia curve. But the error is not very great and becomes less as the piston areas increase.

After determining the effective piston force, the pressures on the crossheads and crank pin were computed. The maximum pressure in pounds per square inch of crank-pin projected area are as follows:

	Top.	Bottom.
H.P.	560	500
I.P.	570	460
F.L.P.	400	410
A.L.P.	400	400

Mean velocity, in feet per minute, of rubbing surface in bearing:

Crank pin	667
Main bearing	616

By the above it will be seen that the crank-pin brasses are subjected to very heavy pressures twice per second—127 R.p.m. is the condition selected for this computation—and the viscosity of some lubricants may not be sufficient to maintain a film between the crank pin and its brasses. If the film breaks down there will result for an instant, perhaps, heating and dragging of the bearing metal. It is also evident from the diagram that no change in the valve settings would cushion the inertia forces, because the inertia is greater than the available steam pressure; in the case of the L.P. cylinders, the crank has revolved an angle of about 50 or 60 degrees before the steam pressure balances the inertia.

The inertia curve for 100 R.p.m. has been plotted because the graphical interpretation is so much easier than the statement that the inertia varies as the square of the revolutions.

MAIN BEARINGS.

The computation for the main bearing pressures is by the following arbitrary method: The shaft is assumed to fit closely in its bearings and slightly elastic so that a light stress will put the shaft against its bearings. Hence, one-half of the effective piston force plus one-half of the resultant, vertical component of the crank-pin and crank-web centrifugal force is assumed to be distributed on each bearing. The resultant force of the valve gear is added algebraically to its cylinder crank-bearing pressure. In the case of a bearing between two cranks, the algebraic sum of one-half the effective piston forces plus the crank-pin and crank-web forces, as above, is taken. The pressure is reduced to the pressure per square inch of projected area of bearing surface. The curves

drawn show the vertical force for the crank angle and not the normal pressure at crank radius. This method does not, of course, take into consideration the torsional effect on the crank shaft or the moments set up by the valve and piston pressures.

For convenience in computing, the following factors were determined and computation forms made:

1 B

2 $\frac{1}{\cos B}$

3 $\sin (A + B)$

4 $\cos (A + B)$

5 $\tan B$

6 $5 - (\cos A + 4 \cos B)$

7 $\cos A$

8 $\cos A + Z_1$ was taken from Foley's Mechanical Engineers' Reference Book, and the values check very closely with those given by Naval Constructor D. W. Taylor, U. S. N., in Volume 9 of the Transactions Society of Naval Architects and Marine Engineers.

9 y

10 $\cos x$

11 $\cos x + \frac{\cos 2x}{E}$ Equation 17 with substitutions.

12 $\frac{\sin (x + y)}{\cos y}$ Equation 5 with substitutions.

13 $19.86 - (\cos x + 18.86 \cos y)$ Equation 9 with substitutions.

H.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Cos A	1.000 00	.984 81	.906 31	.707 11	.422 62	.173 65	.000 00	—.173 65	—.422 62	—.707 11	—.906 31	—.984 81	—1.000 00
$F_v = F \cos A$	39 650	39 000	35 900	28 030	16 750	6 890	0	—6 890	—16 750	—28 030	—35 900	—39 000	—39 650
$F_o = G \pm F_v$	36 040	35 390	32 290	24 420	13 140	3 280	—3 610	—10 500	—20 360	—31 640	—39 510	—42 610	—43 260
Multiplier	1.250 0	1.220 9	1.070 5	.711 4	.260 3	—.068 3	—.258 4	—.415 6	—.584 7	—.702 9	—.742 6	—.749 2	—.750 0
$f_v = f \times$ mult.	131 200	128 200	112 400	74 700	27 330	—7 170	—27 110	—43 650	—61 400	—73 800	—78 000	—78 600	—78 750
$P_R = g \pm f_v$	121 650	118 650	102 850	65 150	17 780	—16 720	—36 660	—53 200	—70 950	—83 350	—87 550	—88 150	—88 300
$P_F = P_R + F_o$	157 690	154 040	135 140	89 570	30 920	—13 440	—40 270	—63 700	—91 310	—114 990	—127 060	—130 760	—131 560
P_{FA}	135.6	132 4	116.1	77.0	26.6	—12.1	—36.3	—57.6	—82.5	—103.8	—114.8	—118.3	—118.9
P_{FA} down	—50	—6	26	62	101	135	157	176	195	200	180	153	118
P_{FA} up	111	158	185	185	153	116	96.0	79.0	65.0	53.0	42.0	29	0
P down	—55 200	—6 650	30 250	72 200	117 500	157 100	182 700	204 800	227 000	232 800	209 400	178 200	137 400
P up	122 800	175 000	210 000	210 000	169 500	128 500	106 300	87 500	72 000	58 700	46 500	32 100	0
$\frac{1}{\cos B}$	1.000 00	1.000 94	1.005 63	1.016 00	1.026 70	1.031 76	1.032 80	1.031 76	1.026 70	1.016 00	1 005 63	1.000 94	1.000 00
$R = \frac{P}{\cos B}$ down	—55 200	—6 650	30 450	73 400	120 700	162 200	188 700	211 200	233 000	236 500	211 000	178 200	137 400
$R = \frac{P}{\cos B}$ up	122 800	175 000	211 000	213 400	174 000	132 600	109 600	90 200	73 800	59 600	46 800	32 100	0
Sin (A + B)	0.000 00	.216 24	.516 01	.820 97	.978 50	.997 25	.968 24	.911 74	.786 98	.570 97	.324 50	.130 73	.000 00
T = R sin (A+B) down	0	—1 440	15 700	60 300	118 100	161 600	182 600	192 600	183 500	135 000	68 500	23 290	0
T = R sin (A+B) up	0	37 850	108 900	175 200	170 200	132 100	106 200	82 300	58 100	34 050	15 170	4 195	0
$K = \frac{R}{430}$ down	—128.4	—15.5	70.8	170.8	280.6	377.0	438.7	491.0	542.0	550.0	490.6	414.5	319.5
$K = \frac{R}{430}$ up	285.3	406.7	490.5	496.0	404.5	308.2	254.9	210.0	171.6	138.5	108.8	74.6	0
Tan B	0.000 00	.043 45	.106 25	.179 61	.232 63	.254 02	.258 20	.254 02	.232 63	.179 61	.106 25	.043 45	000 00
C = P tan B down	0	—289†	3 215	12 960	27 340	39 880	47 200	52 000	52 700	41 800	22 210	7 740	0
C = P tan B up	0	7 560	22 320	37 700	39 400	32 610	27 420	22 210	16 740	10 550	4 950	1 395	0
Ca — down	0	—0.47†	3.5	14.1	29 8	43.4	51.4	56.6	57.4	45.5	24 2	8.4	0
Ca — up	0	8.28	24.3	41.0	41.9	35.5	29.8	24.2	18.2	11.5	5.4	1.5	0
Cos (A + B)	1.000 00	.976 34	.856 59	.570 97	.276 27	.074 16	—.250 00	—.410 76	—.616 98	—.820 97	—.945 89	—.991 42	—1.000 00
N = R cos (A+B) down	—55 200	—6 490	26 100	41 900	24 900	12 020	—47 150	—86 900	—143 700	—194 200	—199 500	—176 500	—137 400
N = R cos (A+B) up	—122 800	—170 700	—180 800	—122 000	—35 900	—9 840	27 400	37 080	45 500	48 950	44 250	31 820	0
Piston displacement from top = 5—(4 cos B+cos A)	0.000 00	.018 95	.116 10	.355 89	.681 42	.949 47	1.127 00	1.296 77	1.526 66	1.770 11	1.928 71	1.988 57	2.000 00
Sine curve	0.000 00	.015 19	.093 69	.292 89	.577 38	.826 35	1.000 00	1.173 65	1.422 62	1.707 11	1.906 31	1.984 81	2.000 00
Difference	0	.003 76	.022 41	.063 00	.104 04	.123 12	.127 00	.123 12	.104 04	.063 00	.022 40	.003 76	0
Angle B	0° 0′ 0″	2° 29′ 16.8″	6° 3′ 53.0″	10° 10′ 56.3″	13° 5′ 45.0″	14° 15′ 9.9″	14° 28′ 39.1″	14° 15′ 9.9″	13° 5′ 45.0″	10° 10′ 56.3″	6° 3′ 53.0″	2° 29′ 16 8″	0° 0′ 0″

† On backing face.

I.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_V = F \cos A$													
*$F_0 = G \pm F_V$													
*Multiplier													
$f_V = f \times$ mult	153 800	150 200	131 800	87 600	32 050	—8 410	—31 800	—51 200	—72 000	—86 500	—91 400	—92 100	—92 300
$P_B = g \pm f_V$	142 610	139 010	120 610	76 410	20 860	—19 600	—42 990	—62 390	—83 190	—97 690	—102 590	—103 290	—103 490
$P_Y = P_B + F_0$	178 650	174 400	152 900	100 830	34 000	—16 320	—46 600	—72 890	—103 550	—129 330	—142 100	—145 900	—146 750
P_{FA}	56.4	55.0	48.3	31.8	10.7	—5.2	—15.0	—23.5	—33.3	—41.6	—45.7	—46.9	—47.2
P_{FR} { down	—14.0	—3 0	6.5	22.0	34.0	45.5	55.0	63.0	72.0	75.0	64.0	51.5	39.0
P_{FR} { up	39.0	50.0	58.0	63.0	58.5	46.0	37.0	28.0	18.0	13.5	10.0	3 0	—13.0
P { down.	—43 550	—9 330	20 580	69 700	107 600	144 000	174 200	199 400	228 000	237 300	202 500	163 000	123 500
P { up	121 300	155 500	180 400	196 000	182 000	143 200	115 200	87 100	56 000	42 000	31 100	9 330	—41 150
* $\frac{1}{\cos B}$													
$R = \frac{P}{\cos B}$ { down	—43 550	—9 340	20 690	70 850	110 400	148 500	180 000	205 800	234 100	241 100	203 500	163 200	123 500
$R = \frac{P}{\cos B}$ { up	121 300	155 600	181 300	199 200	187 000	147 600	119 000	89 900	57 600	42 650	31 300	9 340	—41 150
*Sin (A + B)													
$T = R \sin (A+B)$ { down	0	—2 020	10 670	58 200	108 000	148 000	174 300	187 600	184 200	137 600	66 100	21 300	0
$T = R \sin (A+B)$ { up	0	33 650	93 700	163 500	183 000	147 200	115 200	82 000	45 300	24 350	10 150	1 220	0
$K = \frac{R}{430}$ { down	—101.0	—21.7	48.0	165.0	257.0	345.0	419.0	479.0	544.0	561.0	473.0	379.0	287.0
$K = \frac{R}{430}$ { up	282.0	362.0	422.0	463.0	435.0	343.0	277.0	209.0	134.0	99.0	73.0	21.7	—96.0
*Tan B													
C = P tan B { down	0	—405†	2 190	12 500	25 050	36 360	44 950	50 650	53 100	42 650	21 520	7 080	0
C = P tan B { up	0	6 755	19 160	35 200	42 320	36 350	29 750	22 100	13 020	7 550	3 310	405	0
Ca — { down	0	—0.6†	2.4	13.6	27.3	39.8	49.0	55.2	57.8	46.5	23.4	7.7	0
Ca — { up	0	7.4	20.9	38.3	46.1	39.6	32.4	24.1	14.2	8.2	3.6	0.4	0
*Cos (A + B)													
N = R cos(A+B) { down	—43 550	—9 120	17 720	40 450	22 780	11 010	—45 000	—84 550	—144 500	—198 000	—192 400	—161 600	—123 500
N = R cos(A+B) { up	—121 300	—151 800	—155 400	—113 700	—38 600	—10 940	29 750	36 920	35 550	35 040	29 600	9 260	—41 150
*Piston displacement from top = s—(4 cos B + cos A)													
*Sine curve													
*Difference													
*Angle B													

* The values are the same for each cylinder. † On backing face.

F.L.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_V = F \cos A$	32 400	31 900	29 360	22 900	12 700	5 630	0	—5 630	—13 700	—22 900	—29 360	—31 900	—32 400
*$F_S = G \pm F_V$	29 460	28 960	26 420	19 960	9 760	2 690	—2 940	—8 570	—16 640	—25 840	—32 300	—34 840	—35 340
*Multiplier													
$f_V = f \times$ mult.	141 800	138 600	121 600	80 800	29 550	—7 750	—29 320	—47 200	—66 400	—79 800	—84 400	—85 000	—85 200
$P_R = g \pm f_V$	131 480	128 280	111 280	70 480	19 230	—18 070	—39 640	—57 520	—76 720	—90 120	—94 720	—95 320	—95 520
$P_T = P_R + F_S$	100 940	157 240	137 700	90 440	28 990	—15 380	—42 580	—66 090	—93 360	—115 960	—127 020	—130 160	—130 860
P_{FA}	37.3	36.4	31.9	21.0	6.7	—3.6	—10.0	—15.5	—21.9	—27.2	—29.8	—30 5	—30.7
P_{FR} down	—33.5	—27.0	—19.0	—5.4	10.0	21.0	27.5	33.3	38.5	39.4	37.7	34 4	30.7
P_{FR} up	36.5	41.7	41.5	37.5	28.0	18.3	12.5	7.8	0	—5.0	—11.0	—15.8	—21.5
P down	—144 500	—116 500	—82 000	—23 300	43 150	90 600	118 600	143 600	166 200	170 100	162 700	148 400	132 500
P up	155 400	177 500	176 600	159 700	119 200	77 900	53 200	33 200	0	—21 300	—46 850	—67 300	—91 500
* $\frac{1}{\cos B}$													
R = $\frac{P}{\cos B}$ down	—144 500	—116 600	—82 500	—23 690	44 300	93 500	122 500	148 200	170 500	172 800	163 600	148 500	132 500
R = $\frac{P}{\cos B}$ up	155 400	177 600	177 600	162 200	122 400	80 300	54 900	34 250	0	—21 650	—47 100	—67 400	—91 500
*Sin (A + B)													
T = R sin (A+B) down	0	—25 220	—42 570	—19 440	43 350	93 200	118 500	135 200	134 200	98 700	53 200	19 410	0
T = R sin (A+B) up	0	38 400	91 600	133 200	119 800	80 000	53 200	31 230	0	—12 360	—15 310	—8 810	0
K = $\frac{R}{430}$ down	—336.0	—271.0	—198.0	—55.1	103.1	217.5	285.0	344 2	396.2	408.0	380.5	345.5	308 0
K = $\frac{R}{430}$ up	361.2	413.0	413.0	377.0	284.5	186.8	127.8	79.7	0	—50.3	—109.6	—156.8	—213.0
*Tan B													
C = P tan B down	0	—5 060†	—8 720†	—4 185†	10 040	23 010	30 620	36 450	38 640	30 520	17 280	6 450	0
C = P tan B up	0	7 710	18 760	28 650	27 720	19 770	13 730	8 430	0	—3 825†	—4 985†	—2 925†	0
Ca — down	0	—17.4†	—19.6†	—9.4†	16.4	37.6	50.0	59.5	63.1	49.9	28.2	10 5	0
Ca — up	0	12.6	30.6	46.8	45.3	32.3	22.4	13.8	0	—8.6†	—11.2†	—6.6†	0
*Cos (A + B)													
N = R cos (A+B) down	—144 500	—113 800	—70 700	—13 530	9 150	6 940	3 060	60 900	105 100	141 800	154 700	147 100	132 500
N = R cos (A+B) up	155 400	173 300	152 100	92 600	25 230	5 960	13 720	14 070	0	—17 780	—44 520	—66 750	—91 500
*Piston displacement from top = 5 — (4 cos B + cos A)													
*Sine curve													
*Difference													
*Angle B													

*The values are the same for each cylinder.

†On backing face.

A.L.P. CYLINDER.

Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos A													
*$F_T = F \cos A$													
*$F_0 = G \pm F_T$													
*Multiplier													
*$f_T = f \times$ mult													
*$P_R = g \pm f_T$													
*$F_P = P_R + F_0$													
*P_{FA}													
P_{FA} down	—30.0	—25.0	—16.9	—3.0	11.9	22.0	27.9	32.8	37.8	39 4	36.8	33.3	28 5
P_{FA} up	37.5	40.5	40 0	35.5	26.5	18.5	13.3	8.0	1.5	—4.5	—9.4	—13.3	—18.0
P down.	—129 500	—107 900	—72 900	—12 950	51 400	95 000	120 400	141 500	163 200	170 100	158 800	143 600	123 000
P up	159 700	172 500	170 300	151 100	112 800	78 800	56 600	34 070	6 380	—19 160	—40 020	—56 600	—76 600
*$\frac{1}{\cos B}$													
R = P/cos B down	—129 500	—108 000	—73 300	—13 160	52 800	98 000	124 400	146 000	167 500	172 800	159 600	143 700	123 000
R = P/cos B up	159 700	172 600	171 300	153 500	115 800	81 300	58 500	35 150	6 550	—19 460	—40 270	—56 700	—16 600
*Sin (A + B)													
T = R sin (A+B) down	0	—23 350	—37 810	—10 810	51 700	97 700	120 400	133 200	131 800	98 700	51 900	18 770	0
T = R sin (A+B) up	0	37 300	88 400	126 000	113 400	81 000	56 600	32 050	5 150	—11 110	—13 080	—7 410	0
K = R/430 down	—301.0	—251 0	—170.6	—30 6	122.8	228.0	289.0	339.0	389.0	402.0	371.0	334.0	286 0
K = R/430 up	371.0	401.0	398.0	357.0	269.0	189.2	136.0	81.7	15.2	—45.3	—93 6	—132.0	—178.3
*Tan B													
C = P tan B down	0	—4 690†	—7 750†	—2 325†	11 950	24 120	31 100	35 920	37 940	30 520	16 880	6 240	0
C = P tan B up	0	7 500	18 110	27 120	26 240	20 200	14 620	8 655	1 484	—3 441†	—4 255†	—2 460†	0
Ca — down	0	—10.6†	—17.5†	—5 2†	19.5	39.4	50.8	58.7	62.0	49.9	27.6	10 2	0
Ca — up	0	12.3	29.6	44.3	42.9	33.0	23.9	14.2	2 4	—7.7†	—9.6†	—5.5†	0
*Cos (A + B)													
N = R cos (A+B) down	—129 500	—105 300	—62 800	7 520	10 900	7 270	—31 100	—60 000	—103 300	—141 800	—150 900	—142 300	—123 000
N = R cos (A+B) up	—159 700	—168 400	—146 700	—87 600	—23 900	—6 030	14 620	14 440	4 040	15 980	38 080	56 150	76 600
*Piston displacement from top = s—(4 cos B+cos A)													
*Sine curve													
*Difference													
*Angle B													

* The values are the same for each cylinder.

† On backing face.

H.P. VALVE GEAR.

Angle x	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Cos x	1.000 00	.984 81	.906 31	.707 11	.422 62	.173 65	.000 00	—.173 65	—.422 62	—.707 11	—.906 31	—.984 81	—1.000 00
V = v cos x	3 470	3 418	3 143	2 453	1 466	603	0	—603	—1 466	—2 453	—3 143	—3 418	—3 470
$V_v = l \pm V$	2 030	1 978	1 703	1 013	26	—837	—1 440	—2 043	—2 906	—3 893	—4 583	—4 858	—4 910
Multiplier	1.053 02	1.034 63	.940 39	.707 11	.388 54	.123 83	—.053 02	—.223 47	—.456 70	—.707 11	—.872 23	—.934 99	—.946 98
k = K x mult	5 640	5 540	5 035	3 785	2 080	663	—284	—1 197	—2 445	—3 785	—4 670	—5 010	—5 020
M = H ± k	5 317	5 217	4 712	3 462	1 757	340	—607	—1 520	—2 768	—4 108	—4 993	—5 333	—5 343
p = M + V_v	7 347	7 195	6 415	4 475	1 783	—497	—2 047	—3 563	—5 674	—8 001	—9 576	—10 191	—10 253
sin (x + y)	0.000 00	.182 71	.442 94	.733 64	.926 64	.993 88	1.000 00	.975 73	.885 98	.680 58	.402 30	.164 58	.000 00
$t = \frac{p \sin (x+y)}{\cos y}$ (cos y)	0	1 314	2 840	3 288	1 652	—494	—2 047	—3 480	—5 030	—5 450	—3 850	—1 678	0
$Q = p \frac{m}{r}$	0	288	622	719	362	—108	—448	—762	—1 100	—1 192	—842	—367	0
$E_p = \frac{p}{178}$	41.3	40.5	36.0	25.2	10.0	—2.8	—11.5	—20.0	—31.9	—45.0	—53.8	—57.2	—57.6
Valve displacement from top = 19 86 — (cos x + 18.86 cos y)	0.00	.01	.10	.30	.60	.85	1.03	1.19	1 44	1.72	1.92	1.99	2.00
Sine curve	0.00	.015	.094	.293	.577	.826	1.00	1 174	1.423	1.707	1.906	1 985	2.00
Scale for plotting	0.00	.01	.05	.16	.32	.45	.54	.64	.76	.90	1.01	1.05	1.05
Angle y	0° 0′ 0″	0° 31′ 39.3″	1° 17′ 2.4″	2° 8′ 55.2″	2° 45′ 15.8″	2° 59′ 35.6″	3° 2′ 21.8″	2° 59′ 35.6″	2° 45′ 15 8″	2° 8′ 55·2″	1° 17′ 2.4″	0° 31′ 39.3″	0° 0′ 0″

I.P. VALVE GEAR.

Angle x	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos x													
*$V = v \cos x$													
*$V_v = l \pm V$													
*Multiplier													
$k = K \times$ mult	9 540	9 370	8 520	6 410	3 522	1 123	—481	—2 050	—4 140	—6 410	—7 910	—8 480	—8 580
$M = H \pm k$	9 580	9 410	8 560	6 450	3 562	1 163	—441	—2 010	—4 100	—6 370	—7 870	—8 440	—8 540
$p = M + V_v$	11 610	11 388	10 263	7 463	3 588	—326	—1 881	—4 053	—7 006	—10 263	—12 453	—13 298	—13 450
*$\sin(x + y)$													
$\frac{}{\cos y}$													
$t = \frac{p \sin(x + y)}{\cos y}$	0	2 080	4 540	5 480	3 325	—324	—1 881	—3 955	—6 200	—6 990	—5 010	—2 190	0
$Q = p \frac{m}{r}$	0	455	994	1 198	728	—71	—412	—866	—1 356	—1 528	—1 096	—479	0
$E_p = \frac{p}{178}$	65.2	64.0	57.6	41.9	20.2	—1.9	—10.6	—22.8	—39.3	—57.6	—70.0	—74.7	—75.6
*Valve displacement from top = 19.86 — (cos x + 18.86 cos y)													
*Sine curve													
*Scale for plotting													
*Angle y													

*The values are the same for each cylinder.

F.L.P. AND A.L.P. VALVE GEAR.

Angle x	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
*Cos x													
*V = v cos x													
*V ₁ = l ± V													
*Multiplier													
k = Kx mult	10 670	10 480	9 520	7 160	3 940	1 255	−537	−2 264	−4 625	−7 160	−8 840	−9 480	−9 600
M = H ± k	11 940	11 750	10 790	8 430	5 270	2 525	633	−994	−3 355	−5 890	−7 570	−8 210	−8 330
p = M + V_T	13 970	13 728	12 493	9 443	5 236	1 688	−807	−3 037	−6 261	−9 783	−12 153	−13 068	−13 240
*sin (x + y)													
*$\frac{\cos y}{\cos y}$													
$t = \frac{p \sin (x + y)}{\cos y}$	0	2 510	5 520	6 930	4 860	1 678	−807	−2 963	−5 550	−6 660	−4 890	−2 150	0
$Q = p \frac{m}{r}$	0	550	1 207	1 516	1 063	367	−177	−649	−1 215	−1 456	−1 070	−471	0
$E_p = \frac{p}{178}$	78.5	77.1	70.1	53.1	29.4	9.5	4.5	17.1	35.2	54.9	68.3	73.4	74.3
*Valve displacement from top = 19.86 − (cos x + 18.86 cos y)													
*Sine curve													
*Scale for plotting													
*Angle y													

*The values are the same for each cylinder.

H.P., I.P., F. AND A.L.P.——BACKING VALVE GEAR.

	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
U = q x mult	1 648	1 616	1 470	1 105	608	194	−83	−350	−714	−1 105	−1 364	−1 462	−1 481
o = h ± U	998	966	820	455	−42	−456	−733	−1 000	−1 364	−1 755	−2 014	−2 112	−2 131
O = o + V_T	3 028	2 944	2 523	1 468	−16	−1 293	−2 173	−3 043	−4 270	−5 648	−6 597	−6 970	−7 041
$t_o = \frac{o \sin (x + y)}{\cos y}$	0	538	1 118	1 077	−15	−1 285	−2 173	−2 966	−3 782	−3 850	−2 652	−1 147	0
$Q_o = O \frac{m}{r}$	0	118	244	236	−3	−281	−476	−649	−828	−842	−580	−251	0

MAIN BEARING PRESSURES.

Downward forces are +
Upward forces are —

Pressures in pounds.
Projected bearing area is used.

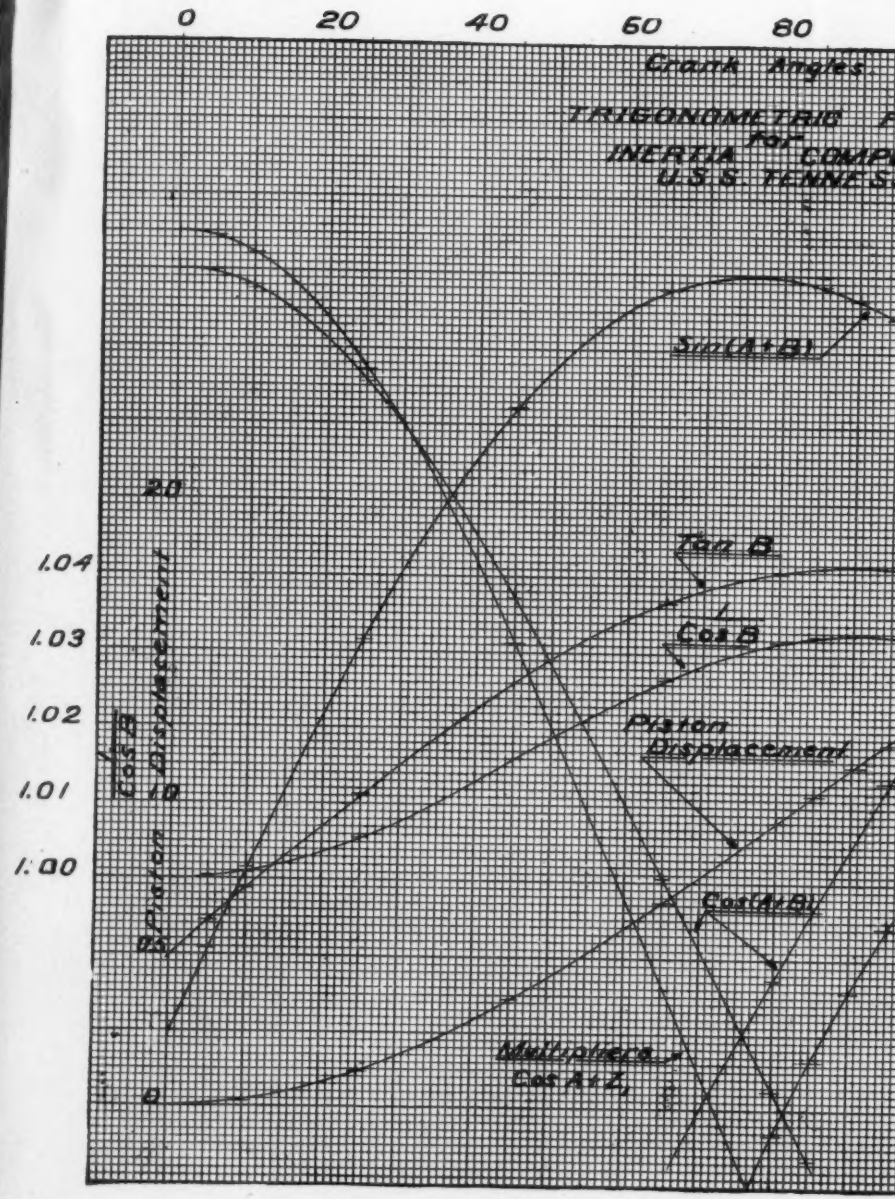
Angle A	0 360	10 350	25 335	45 315	65 295	80 280	90 270	100 260	115 245	135 225	155 205	170 190	180
Main Bearing No. 1:													
$\frac{P}{2}$ F.L.P. down	−72 250	−58 250	−41 000	−11 650	21 575	45 300	59 300	71 800	83 100	85 050	81 350	74 200	66 250
$\frac{P}{2}$ F.L.P. up	−77 700	−88 750	−88 300	−79 850	−59 600	−38 950	−26 600	−16 600	0	10 650	23 425	33 650	45 750
$\frac{S}{2}$	−37 750	−37 150	−34 200	−26 700	−15 970	−6 560	0	6 560	15 970	26 700	34 200	37 150	37 750
S	−34 380	−33 730	−30 770	−23 270	−12 540	−3 130	3 430	9 990	19 400	30 130	37 630	40 580	41 180
$\frac{S}{2}$	−17 160	−16 860	−15 385	−11 635	−6 270	−1 565	1 715	4 995	9 700	15 065	18 815	20 290	20 590
Valve gear, F.L.P. down	14 200	15 000	15 500	14 700	12 800	10 500	8 700	6 800	3 600	−900	−5 300	−8 100	−9 700
Valve gear, F.L.P. up	14 200	13 000	10 700	6 700	1 600	−2 000	−4 700	−6 800	−9 400	11 300	−11 600	−10 600	−9 700
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, down	−75 210	−60 110	−40 890	−8 585	28 110	54 240	69 720	83 600	96 400	99 220	94 870	86 390	77 140
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, up	−80 660	−92 610	−92 990	−84 790	−64 270	−42 520	−29 590	−18 410	300	14 420	30 640	43 340	56 640
Pressure per square inch of bearing, down	−179.5	−143.5	−97.6	−20.5	66.1	129.5	166.3	199.6	230.0	236.8	226.5	206.2	184.0
Pressure per square inch of bearing, up	−192.6	−221.0	−222.0	−202.5	−153.2	−101.5	−70.6	−43.9	0.7	34.4	73.1	103.5	135.2
Main Bearing No. 2:													
$\frac{P}{2}$ H.P. + $\frac{P}{2}$ F.L.P. + S down	21 580	33 760	41 980	50 180	62 180	65 380	68 180	66 880	57 330	39 980	19 830	3 780	−5 570
$\frac{P}{2}$ H.P. + $\frac{P}{2}$ F.L.P. + S up	8 280	−4 870	−20 220	−16 520	1 780	10 980	9 580	4 980	−14 425	−37 570	−60 820	−70 870	−68 820
Pressure per square inch of bearing, down	52.5	82.1	102.2	122.0	151.3	159.1	166.0	162.8	139.3	97.3	48.3	9.2	−13.6
Pressure per square inch of bearing, up	20.2	−11.8	−49.2	−40.2	4.3	26.7	23.3	12.1	−35.1	−91.4	−148.0	−172.5	−167.4
Main Bearing No. 3:													
$\frac{P}{2}$ H.P. down	−27 600	−3 325	15 125	36 100	58 750	78 550	91 350	102 400	113 500	116 400	104 700	89 100	68 700
$\frac{P}{2}$ H.P. up	−61 400	−82 500	−105 000	−105 000	−84 750	−64 250	−53 150	−43 750	−36 000	−29 350	−23 250	−16 050	0
Valve gear down	−4 000	−4 400	−4 800	−4 100	−2 700	−1 000	300	1 800	4 300	7 400	9 800	11 100	11 800
Valve gear up	−4 000	−3 200	−1 800	400	3 400	5 600	6 900	8 200	9 800	11 500	12 200	12 200	11 800
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, down	−48 760	−24 590	−5 060	20 370	49 780	75 990	93 370	109 200	127 500	138 900	133 300	120 500	110 100
$\frac{P}{2}+\frac{S}{2}$ + valve gear, total, up	−82 560	−102 600	−122 190	−116 200	−87 620	−60 280	−44 540	−30 560	−16 500	−2 785	7 765	16 440	32 390
Pressure per square inch of bearing, down	−116.3	−58.7	−12.1	48.6	118.9	181.5	223.0	260.3	304.0	331.5	318.0	287.4	262.5
Pressure per square inch of bearing, up	−197.0	−244.7	−291.3	−277.0	−209.0	−143.7	−106.4	−72.9	−39.4	−6.7	18.5	39.2	77.3

MAIN BEARING PRESSURES—Continued.

Downward forces are +
Upward forces are —

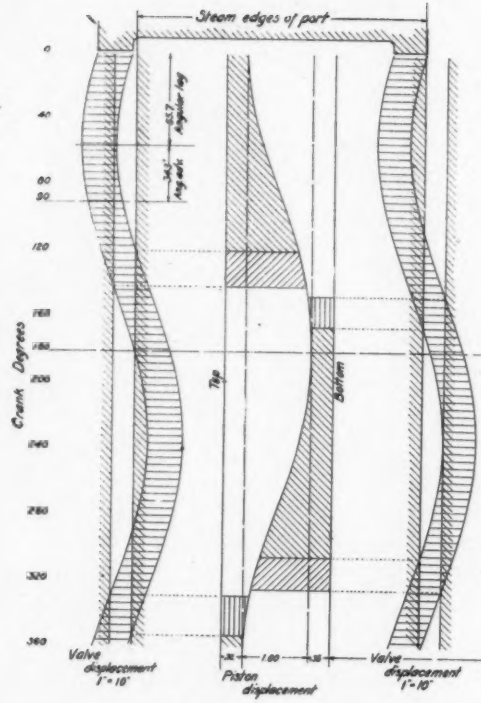
Pressures in pounds.
Projected bearing area is used.

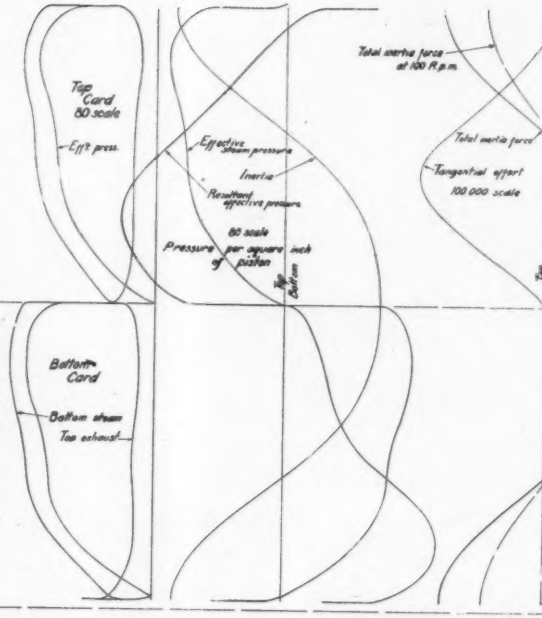
Main Bearing No. 4:													
$\frac{P}{2}$ I.P. { do wn	—21 775	—4 665	10 290	34 850	53 800	72 000	87 100	99 700	114 000	118 650	101 250	81 500	61 750
up	—60 650	—77 750	—90 200	—98 000	—91 000	—71 600	—57 600	—43 550	—28 000	—21 000	—15 550	—4 665	20 575
Valve gear, { down	14 000	14 700	15 300	14 800	12 800	10 900	9 400	7 900	5 200	1 200	—2 600	—5 300	—6 700
up	14 000	12 800	10 700	6 900	2 500	—700	—2 700	—4 500	—6 700	—8 200	—8 500	—7 700	—6 700
$\frac{P}{2}+\frac{S}{2}$ + valve { down	—24 940	—6 825	10 210	38 020	60 330	81 340	98 220	112 600	128 900	134 900	117 500	96 490	75 640
gear, total, up	—63 810	—81 810	—94 890	—108 700	—94 770	—73 870	—58 590	—43 060	—25 000	—14 140	—5 235	7 925	34 470
Pressure per square inch { down	—59.5	—16.3	24.4	90.7	143.9	194.0	234.1	268.7	307.5	321.8	280.3	230.3	180.5
of bearing, up	—152.2	—195.3	—226.3	—245.0	—226.2	—176.4	—139.8	—102.8	—59.6	—33.7	—12.5	18.9	82.2
Main Bearing No. 5:													
$\frac{P}{2}$ I.P. + $\frac{P}{2}$ { down	19 960	27 070	33 730	47 860	54 040	58 400	62 230	63 730	61 030	46 530	19 530	—1 320	—14 670
A.L.P. + S up	4 280	—2 520	—7 370	—9 520	—5 970	2 580	6 030	7 380	1 130	—24 050	—48 570	—55 190	—40 750
Pressure per square inch { down	31.0	42.1	52.5	74.5	84.1	90.8	96.8	99.1	95.0	72.4	30.4	—2.1	—22.7
of bearing, up	6.7	—3.9	—11.5	—14.8	—9.3	4.0	9.4	11.5	1.8	—37.4	—75.5	—85.8	—63.4
Main Bearing No. 6:													
$\frac{P}{2}$ A.L.P. { down	—64 750	—53 950	—36 450	—6 475	25 700	47 500	60 200	70 750	81 600	85 050	79 400	71 800	61 500
up	—79 850	—86 250	—85 150	—75 550	—56 400	—39 400	—28 300	—17 035	—3 190	9 580	20 010	28 300	38 300
Valve gear { down	14 500	15 300	15 700	14 900	12 800	10 600	8 800	6 900	3 800	—700	—5 200	—8 000	—9 400
up	14 500	13 300	10 800	6 800	1 900	—2 000	—4 700	—6 700	—9 300	—11 300	—11 500	—10 500	—9 400
$\frac{P}{2}+\frac{S}{2}$ + valve { down	—67 410	—55 510	—36 140	—3 100	32 230	56 340	70 720	82 650	95 100	99 420	93 020	84 090	72 690
gear, total, up	—82 510	—89 810	—89 740	—80 390	—60 770	—42 970	—31 290	—18 740	—2 790	13 350	27 330	38 090	49 490
Pressure per square inch { down	—161.0	—132.5	—86.2	—7.7	76.9	135.0	168.8	197.3	227.0	237.2	222.0	201.0	173.5
of bearing, up	—197.0	—214.5	—214.2	—192.0	—145.1	—102.5	—74.7	—44.7	—6.7	31 9	65.2	90.9	118.1

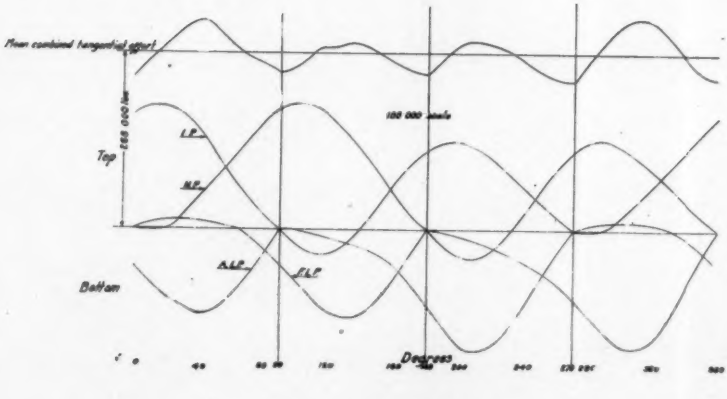

0
20
40
60
80
Crank Angles
TRIGONOMETRIC
INERTIA
U.S.S. TENNE
Sum(A+B)
20
1.04
1.03
1.02
1.01
1.00
Tan B
Cos B
Piston Displacement
Cos(A+B)
Multiplier
Cos A+Z
B

80
100
120
140
160
180
Angles.
TRIG FUNCTIONS
OF
COMPUTATIONS
TENNESSEE
Sin (A+B) and Cos (A+B)
Tan B
Multipliers.
1.3
1.2
1.1
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
E. H. Kenney.
W. B. Robuck.
5 - 5 - '07
PUBLISHED BY A. D. MACLACHLAN, BOSTON.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY.

H. P. CYLINDER







Combined Tangential Efforts

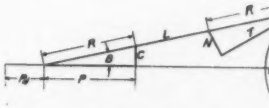
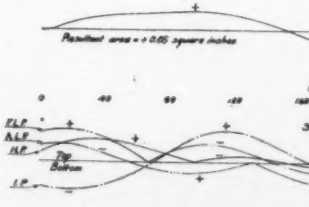
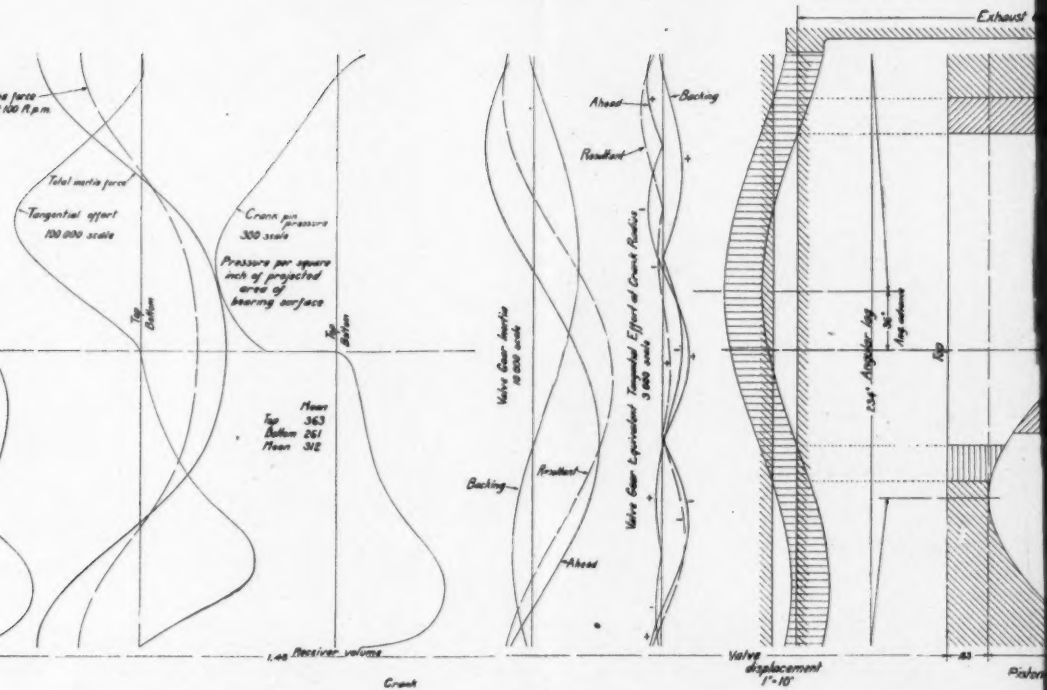
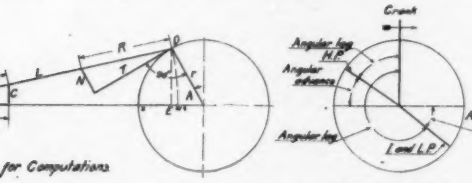


Diagram for Computations



Valve Gear Combined Equivalent Tan





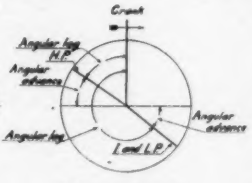
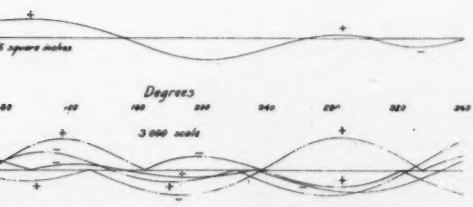
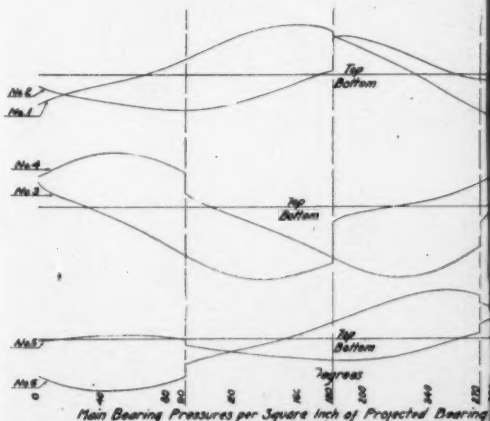


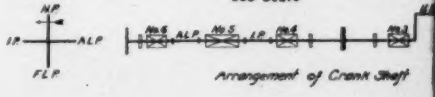
Diagram to Determine Valve Phase.



Combined Equivalent Tangential Efforts at Crank Radius

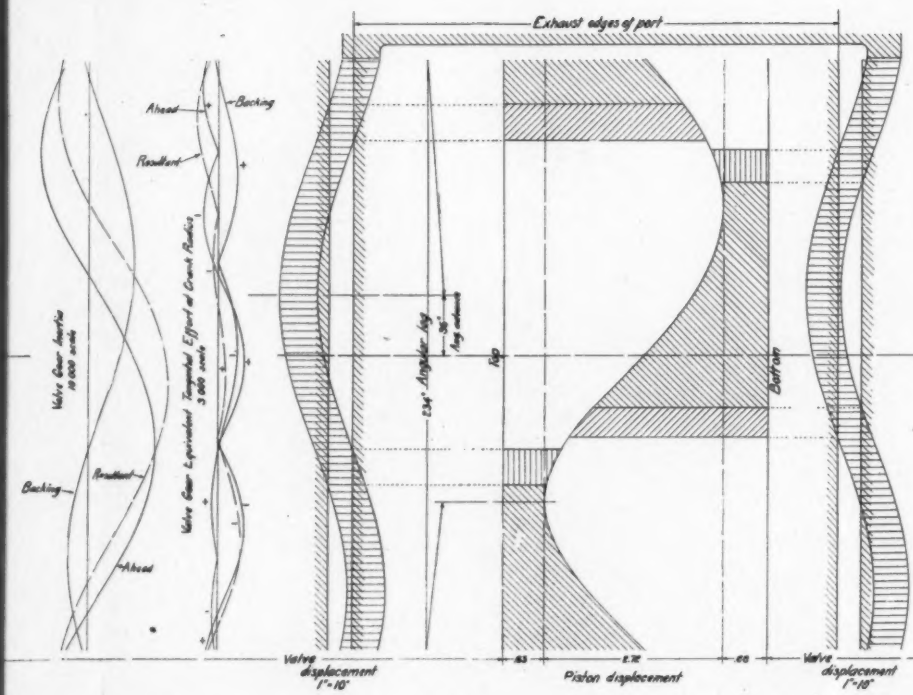


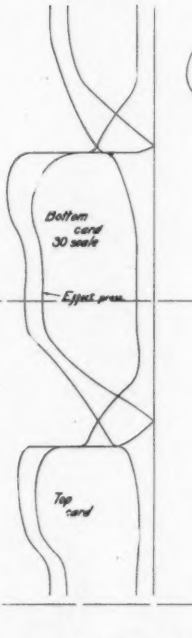
Main Bearing Pressures per Square Inch of Projected Bearing
300 Scale



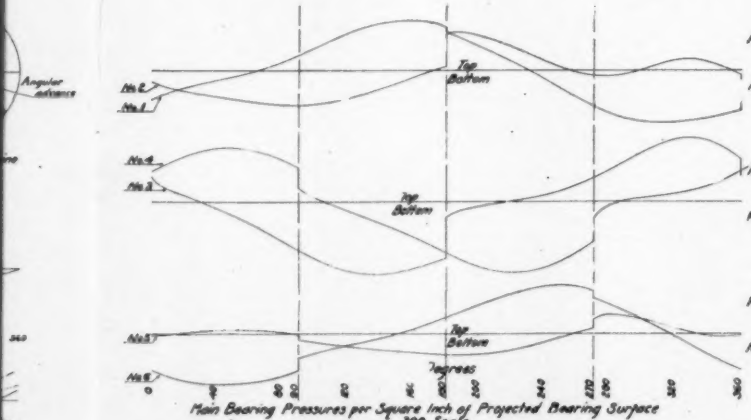
Arrangement of Crank Shaft

I.P. CYLINDER



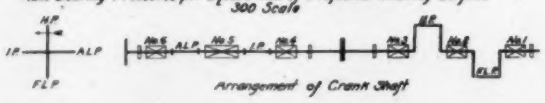


Card no.7 part engine
4-hour trial
R.p.m. 127.0
Knots 22.08



	Top	Bottom
No.1 Mean	127.6	142.4
Mean	135.0	
No.2 Mean	68.7	81.1
Mean	74.9	
No.3 Mean	137.0	174.0
Mean	155.5	
No.4 Mean	128.6	186.6
Mean	157.6	
No.5 Mean	33.3	58.3
Mean	45.8	
No.6 Mean	116.3	139.0
Mean	127.7	

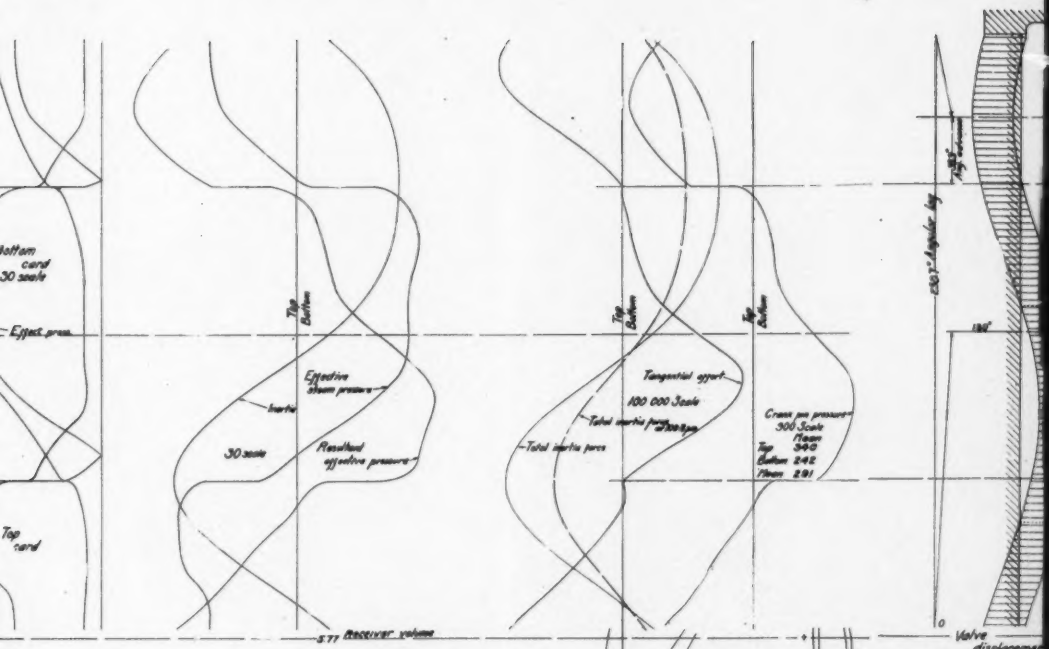
Main Bearing Pressures per Square Inch of Projected Bearing Surface
300 Scale



Arrangement of Crank Shaft

INERTIA AN
P
U.S.
OFFICE OF IN
WM CRAMP AND

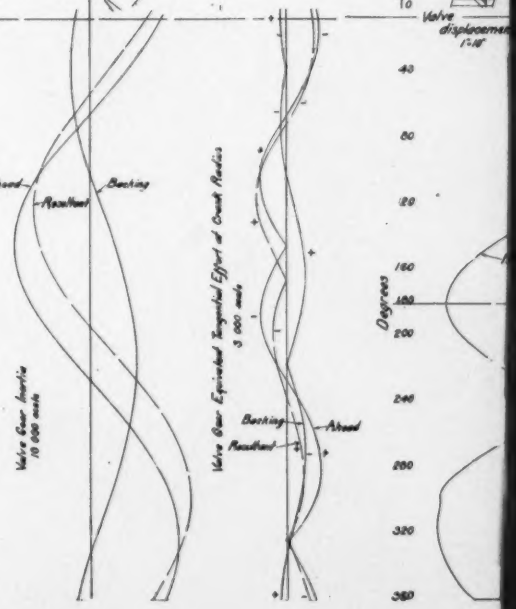
CYLINDER



rd no.7 port engine
4-hour trial
R.p.m. 127.0
Knots 22.08

Indicated Horse Power

H.P.	4 310
I.P.	4 093
F.L.P.	2 239
A.L.P.	2 337
Total	12 979



INERTIA AND TORTIONAL STRESSES
PORT MAIN ENGINE

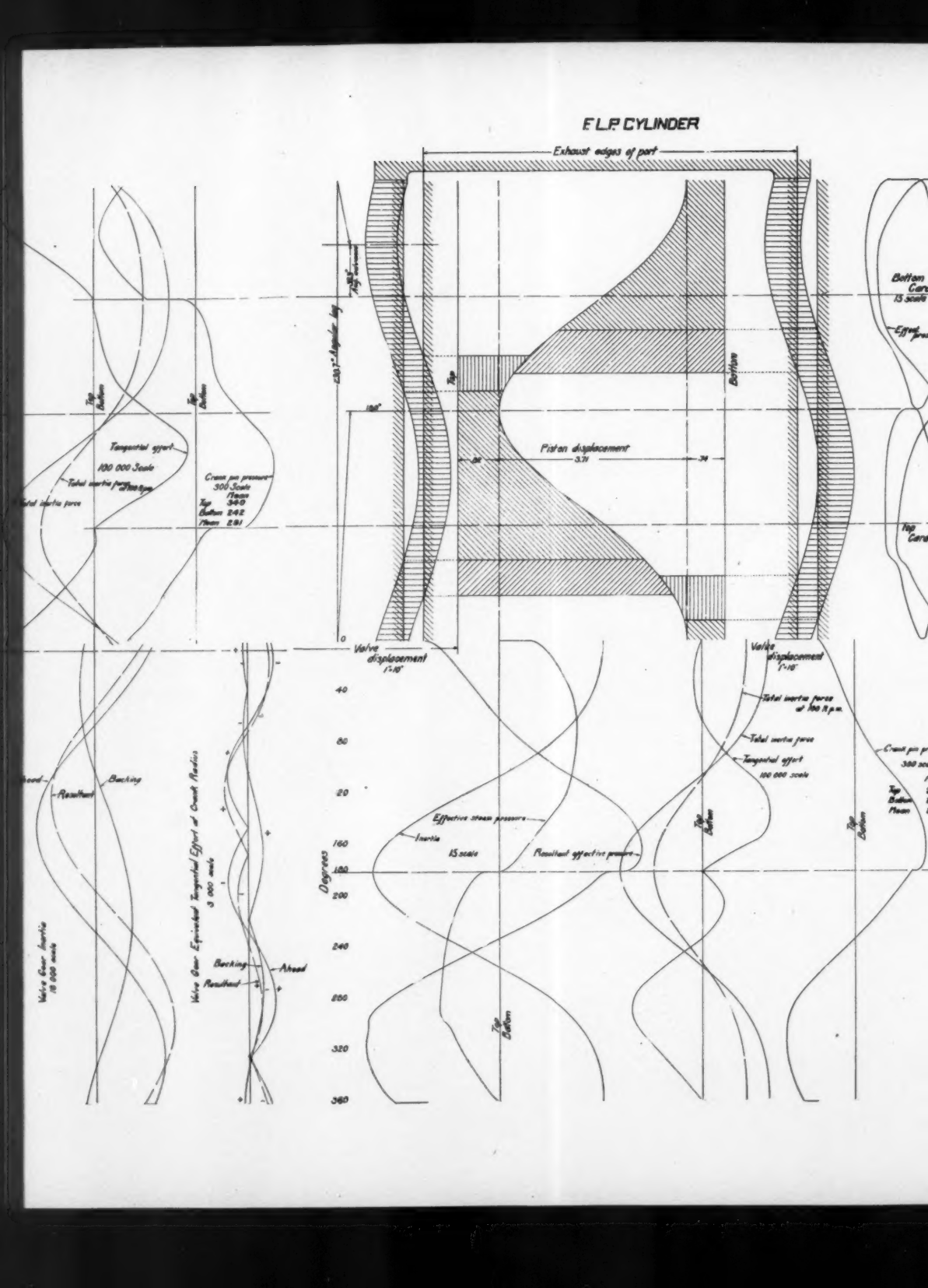
U.S.S. TENNESSEE

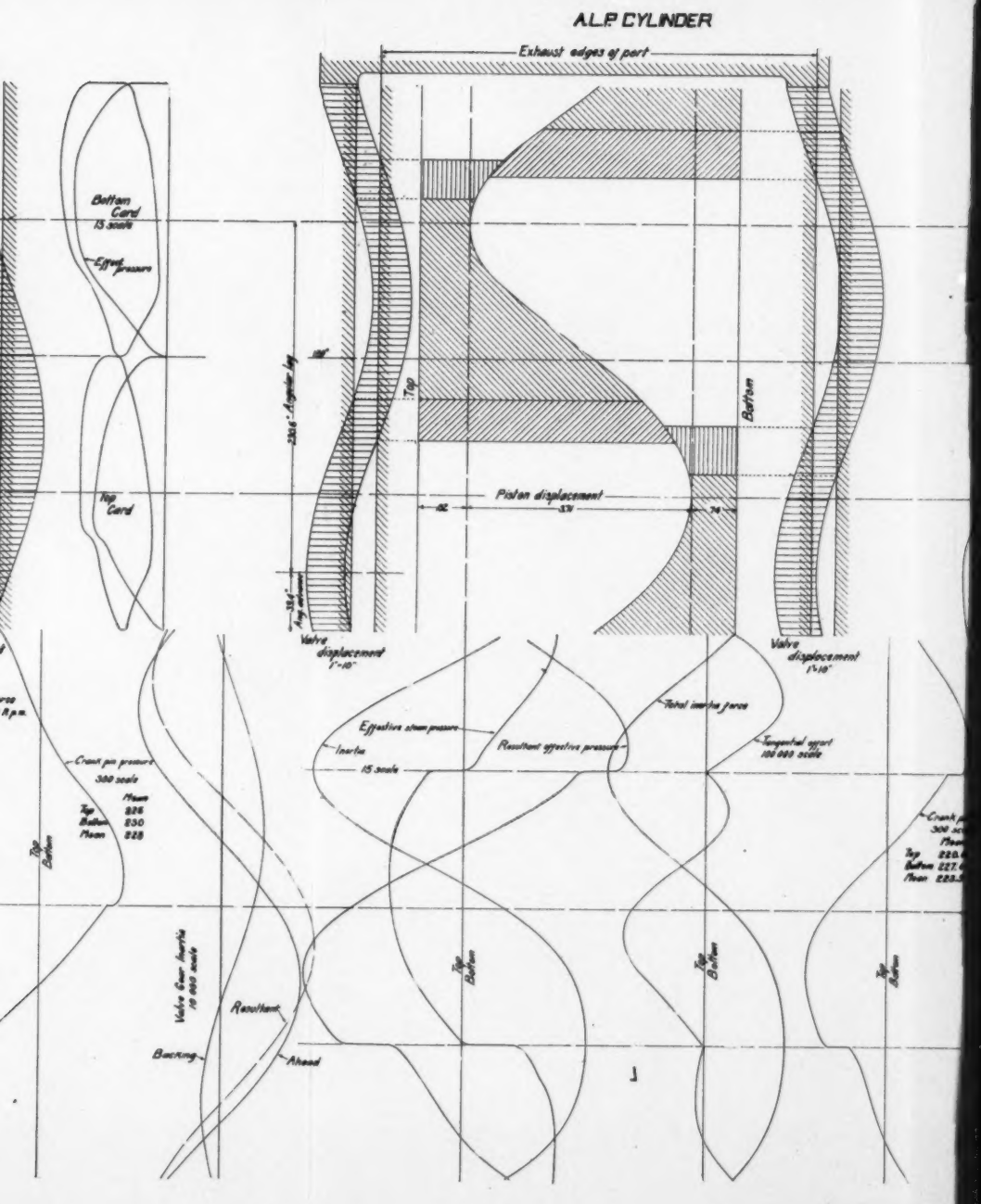
OFFICE OF INSPECTOR OF MACHINERY, U.S.N.

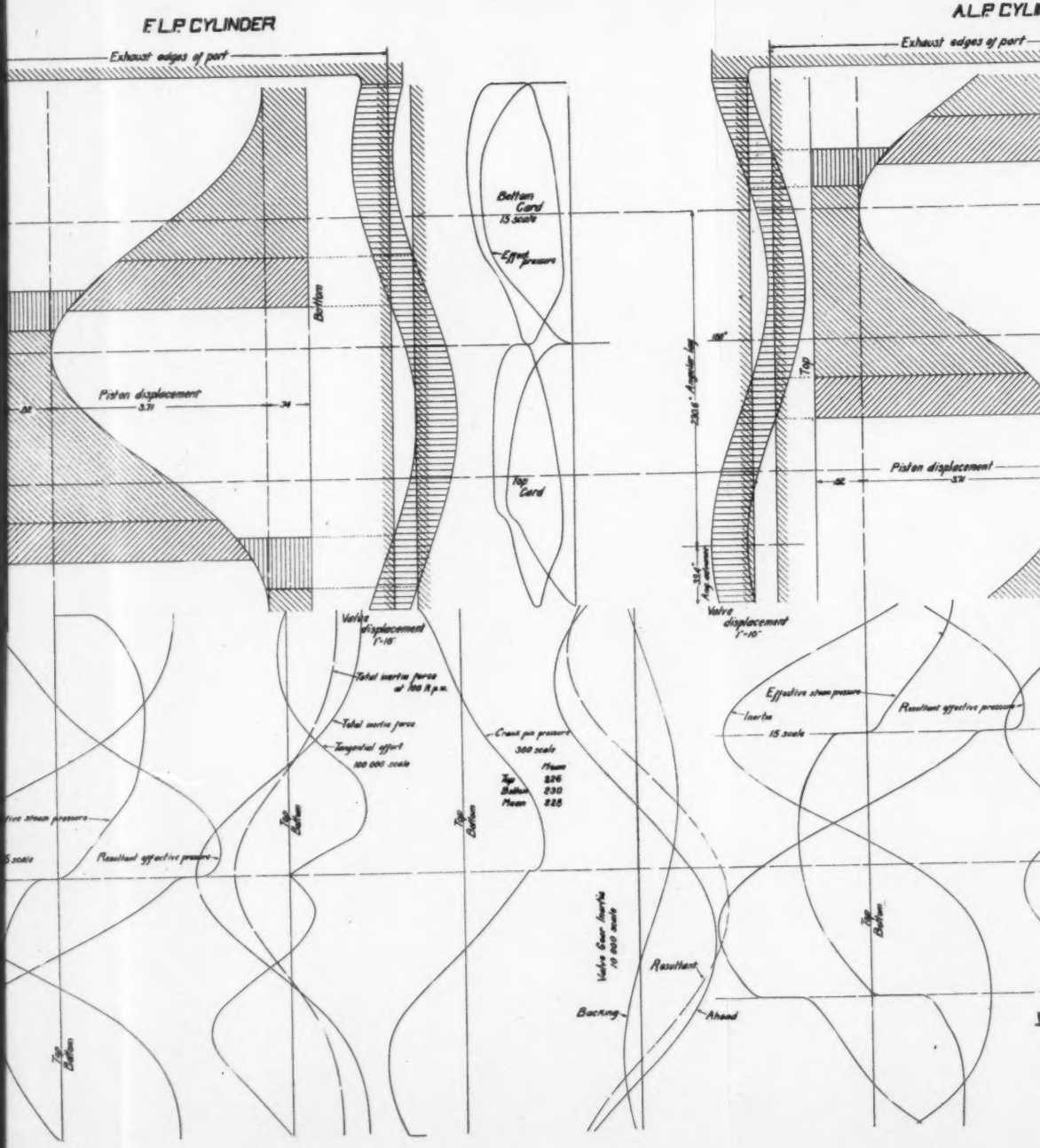
WM CRAMP AND SONS S. AND E.B. CO., PHILA., PA.

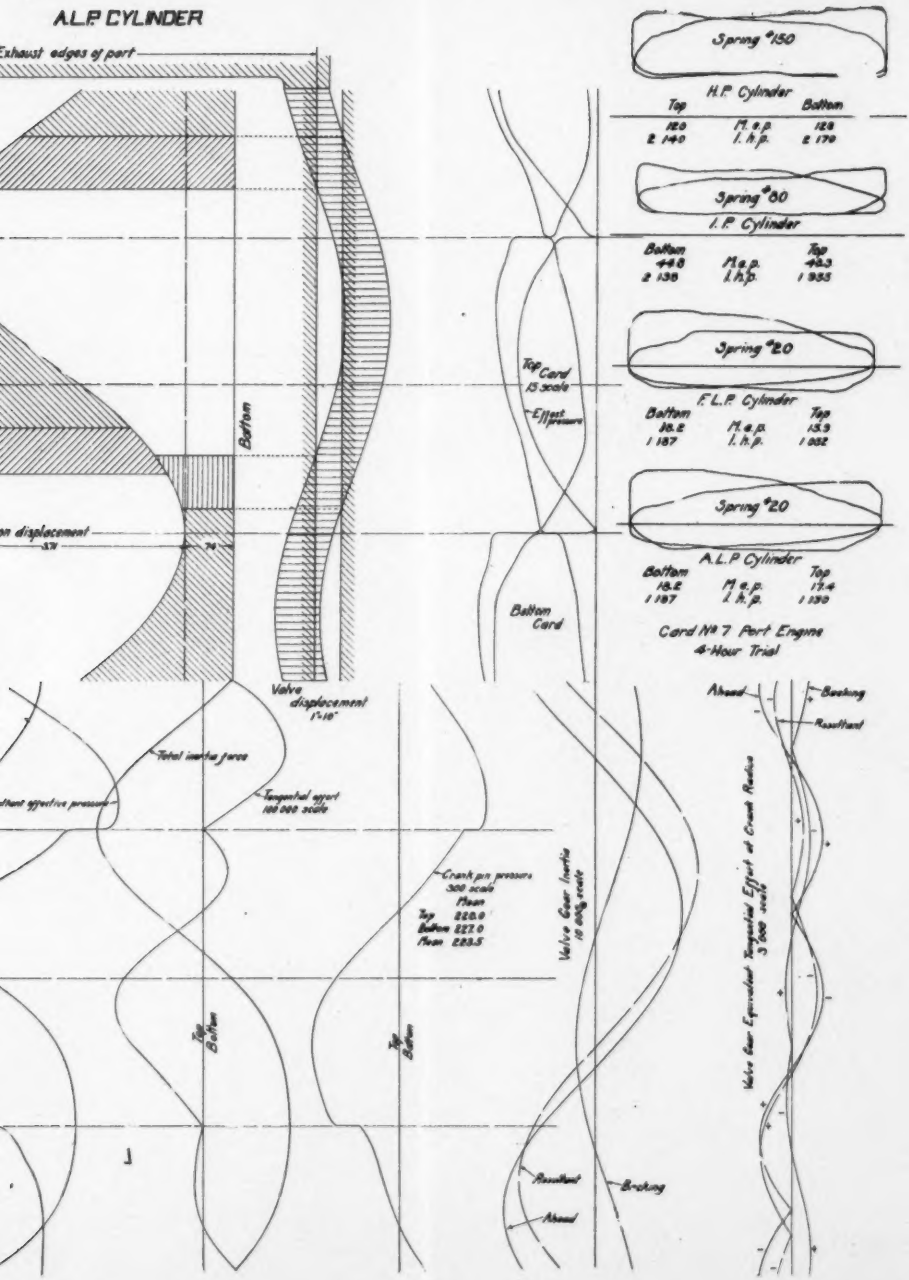
JULY 4, 1907.

Lewis H. Kenney Wm B. Robins

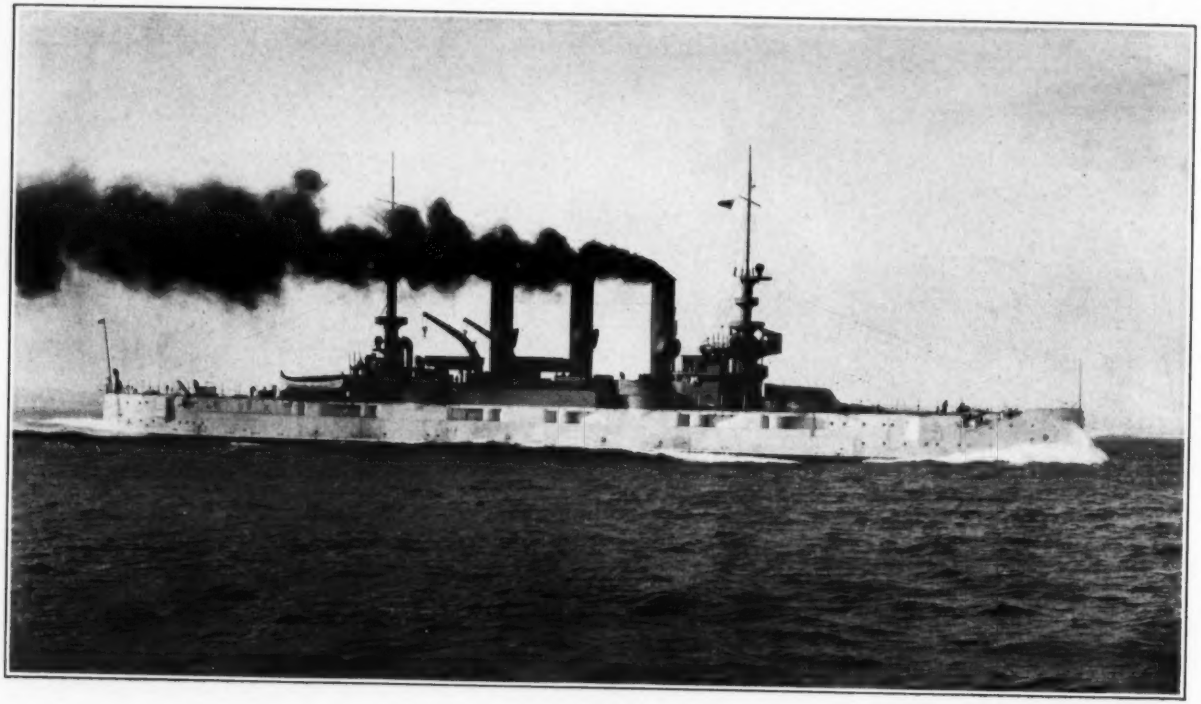

F.L.P. CYLINDER
Exhaust edges of port
Piston displacement
3.71
Valve displacement
Tangential effort
100 000 Scale
Total inertia force
Crank pin pressure
300 Scale
Mean
Top 340
Bottom 242
Mean 291
Effective steam pressure
Inertia
15 scale
Resultant effective pressure
Total inertia force at 100 R.p.m.
Valve Gear Inertia
15 000 scale
Valve Gear Equivalent Tangential Effort at Crank Radius
3 000 scale
Ahead
Backing
Resultant
Degrees
0
40
80
120
160
180
200
240
280
320
360
Top
Bottom
Mean


A.L.P. CYLINDER
Exhaust edges of port
Bottom Card
15 scale
Effort pressure
Top Card
Top
Bottom
Piston displacement
Valve displacement
Valve displacement
Effective steam pressure
Inertia
Resultant effective pressure
Total inertia force
Tangential effort
Crank pin pressure
300 scale
Mean
Top 826
Bottom 830
Mean 828
Valve Gear Inertia
Resultant
Backing
Ahead


F.L.P. CYLINDER
Exhaust edges of port
Bottom
Piston displacement
Bottom Card
15 scale
Effort pressure
Top Card
Valve displacement
Total inertia force at 100 R.p.m.
Total inertia force
Tangential effort
100 000 scale
Crank pin pressure
300 scale
Top
Bottom
Mean
Resultant effective pressure
Valve Gear Inertia
10 000 scale
Resultant
Backing
Ahead
A.L.P. CYLI
Exhaust edges of port
Top
Piston displacement
Valve displacement
Effective steam pressure
Resultant effective pressure
Inertia
15 scale


A.L.P. CYLINDER
Exhaust edges of port
Bottom
Valve displacement 1"=1"
Total inertia force
Tangential effort 100 000 scale
Crank pin pressure 300 scale
Mean
Top 222.0
Bottom 227.0
Mean 223.5
Valve Gear Inertia 10 000 scale
Top
Bottom
Resultant
Backing
Ahead
Spring #150
H.P. Cylinder
Top
Bottom
120
2 140
M.e.p.
I.h.p.
128
2 170
Spring #80
I.P. Cylinder
Bottom
44.8
2 138
Top
49.3
1 935
Top Card 15 scale
Effort pressure
Spring #20
F.L.P. Cylinder
Bottom
18.2
1 187
Top
15.9
1 052
Spring #20
A.L.P Cylinder
Bottom
18.2
1 187
Top
17.4
1 130
Bottom Card
Card No 7 Port Engine 4-Hour Trial
Backing
Resultant
Valve Gear Equivalent Tangential Effort at Crank Radius 3 000 scale

XU



U. S. S. "CONNECTICUT."

U. S. S. *CONNECTICUT.*

OFFICIAL TRIAL.

The *Connecticut* is one of two battleships authorized by Act of Congress, July 1st, 1902, the sister ship being the *Louisiana*, built by the Newport News Shipbuilding & Dry Dock Company, at Newport News, Va. For a full description of this latter vessel, which applies equally to the *Connecticut*, see Volume XVIII, page 171.

STANDARDIZATION TRIAL.

The standardization runs were made over the measured-mile course off Rockland, Me., August 7th, 1907. The weather was cloudy to clear, with a light breeze from South. Smooth sea.

Previous to the vessel starting on the standardization runs, the draught was taken and found to be—

Forward, feet and inches, . . .	24–00
Aft, feet and inches, ...	26–01
Mean, feet and inches, ...	25–00½
Corresponding displacement, tons, .	16,440

These runs gave the data from which the curves on Plate I were plotted.

The results of the runs with and against the tide having been plotted as separate curves, the curve of true speed was obtained, from which it was deduced, from a mean of both engines, that 120.3 revolutions per minute would be required for a true speed of 18 knots.

After the standardization trial was completed, and the vessel was again at anchor, the draught was found to be—

Forward, feet and inches, . . .	23–09½
Aft, feet and inches, ...	26–00
Mean, feet and inches, ...	24–10¾
Corresponding displacement, tons, .	16,300

FOUR-HOURS' FULL-POWER TRIAL.

The four-hours' full-power trial was held on August 9, 1907. The weather was clear with a light northerly breeze. Smooth sea.

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*]

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	263.5	
Mean steam pressure at engines, pounds		
H.P. steam-chest, gauge	224.0	228.0
1st receiver (absolute), pounds	106.0	108.0
2d receiver (absolute), pounds	37.0	29.6
Vacuum in condensers, inches of mercury, mean	21.0	20.9

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	63	63
Discharge, degrees	112	118
Hotwell, degrees	90	90
Feed water, degrees	152	153
Engine room, working platform, degrees	87	94
Firerooms, working level, degrees	93	
Smoke stacks, average, degrees	571	

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

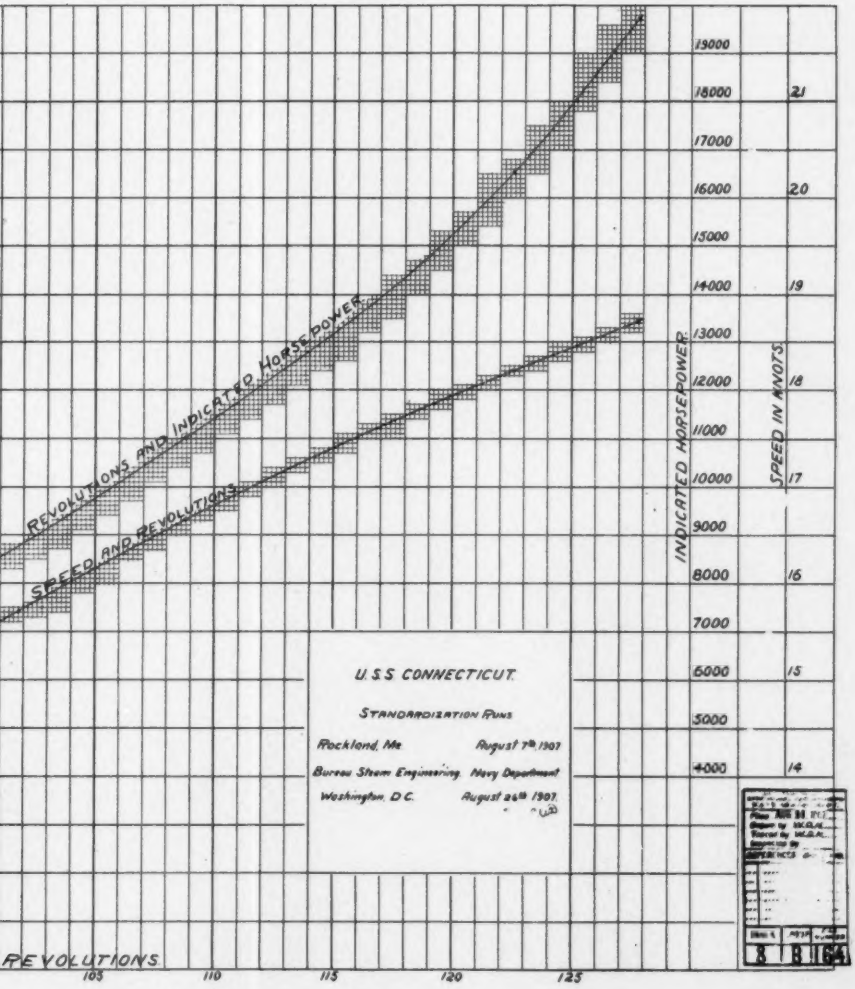
Average revolutions, main engines, per minute	128.48	128.04
Mean revolutions, both engines, per minute	128.26	
Pumps, main air	25.0	26.8
circulating	266.5	211.5
feed, d.s., per minute, aft	37.4	35.0
forward	42.3	38.8
fire and bilge	29.3	
Auxiliary condenser, air and circulating	42.3	20.4
Blower engines	543.5	
Speed of ship, in knots per hour	18.783	
Slip of propeller, in per cent. of its own speed, based on mean pitch	15.94	15.76
Air pressure in firerooms, in inches of water, mean	1.36	

Mean Effective Pressures in Cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	82.38	85.61
I.P. cylinder	43.53	51.92
F.L.P. cylinder	26.68	23.71
A.L.P. cylinder	24.62	21.51
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	53.53	53.59

XU

REVOLUTIONS
SPEED AND
REVOLUTIO
85
90
95
100
105


REVOLUTIONS AND INDICATED HORSE POWER
SPEED AND REVOLUTIONS
INDICATED HORSE POWER
SPEED IN KNOTS
19000
18000
17000
16000
15000
14000
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
21
20
19
18
17
16
15
14
REVOLUTIONS
105
110
115
120
125
U.S.S. CONNECTICUT
STANDARDIZATION RUNS
Rockland, Me. August 7th 1907
Bureau Steam Engineering, Navy Department
Washington, D.C. August 24th 1907

INDICATED HORSEPOWER.

	Starboard.	*Port.*
Main engines, H.P. cylinder	2,080	2,149
I.P. cylinder	2,963	3,523
F.L.P. cylinder	2,412	2,136
A.L.P. cylinder	2,234	1,836
total	9,689	9,644
Collective H.P. of both main engines	19,333	
Air, circulating, feed and hotwell pumps	486	
All other auxiliaries in operation	706	
Total all machinery	20,525	

DEDUCED DATA.

I.H.P. (total) per square foot of grate surface	18.74
Cooling surface (main condenser), square feet per I.H.P. (main engines)	1.072
Heating surface, square feet per I.H.P. (total)	2.57

NOTES.

BABCOCK & WILCOX BOILERS IN THE ROYAL NAVY.

The first of H. M. vessels to be supplied with Babcock & Wilcox patent water-tube boilers was the torpedo gunboat *Sheldrake,* having four boilers, each two of which were placed back to back in separate compartments, and fired fore and aft.

These boilers were made under rigorous survey by Admiralty inspectors, in accordance with the terms of the contract, and one of the four was erected at the constructors' works and there subjected to a number of tests to determine its efficiency.

TRIAL OF H. M. S. *SHELDRAKE.*

At the expiration of the usual basin and sea-going trials the Admiralty decided to test the boilers under actual sea-going conditions, and accordingly an exhaustive series of trials—nine in number—were carried out, each over a distance of 1,000 miles.

The results obtained on these, and the preceding basin and commissioning trials, were as follows:

Date.	Nature of trial.	No. of boilers in use.	No. of boilers in reserve.	Duration in hours.	Steam pressure.	Indicated horsepower.	Air pressure in stokehold.	Coal per I.H.P., all purposes.	Coal per square foot grate.	Water per pound coal.	Ashes per pound coal.	Grate surface.
14-11-98	Evaporative	2	2	8	168	1,116	.0	1.69	15.0	10.5	...	126
15-11-98	Evaporative	2	2	8	179	1,292	.0	1.46	15.0	10.5	...	126
16-11-98	Evaporative	2	2	8	169	1,761	...	1.78	25.0	9.05	...	126
17-11-98	Evaporative	2	2	8	165	1,873	...	1.67	25.0	9.34	...	126
28-11-98	8 hours at 2,500 I.H.P.	4	...	8	152	2,742	.0	1.43	15.0	...	...	252
1-12-98	3 hours at 3,000 I.H.P.	4	...	3	151	4,050	.43	1.57	25.6	...	...	252
22- 2-99	3 hours commissioning	4	...	3	119	2,735	.14	1.64	17.8	...	...	252
28- 2-99	1,000 miles at 1,500 I.H.P.	3	1	69	120	1,303	.0	1.61	12.8	...	.152	189
9- 3-99	1,000 miles at 1,500 I.H.P.	...	1	68	120	1,506	.0	1.6	12.67	...	.150	189
28- 3-99	1,000 miles at 1,500 I.H.P.	3	1	70	135	1,534	.0	1.75	14.2	...	.200	189
20- 4-99	1,000 miles at 1,500 I.H.P.	3	1	68½	130	1,539	.0	1.59	13.1	...	.150	189
5- 5-99	1,000 miles at 1,800 I.H.P.	3	1	67	135	1,829	.0	1.6	15.4	...	.216	189
19- 5-99	1,000 miles at 1,800 I.H.P.	3	1	66½	140	1,838	.0	1.68	16.4	...	.220	189
15- 6-99	1,000 miles at 2,000 I.H.P.	3	1	59	145	2,033	.0	1.57	17.0	...	.220	189
3- 7-99	1,000 miles at 2,000 I.H.P.	3	1	61½	140	2,042	0	1.56	16.8	...	.230	189
20- 7-99	1,000 miles at 2,250 I.H.P.	3	1	56½	150	2,245	.0	1.63	19.4	...	.250	189
Total		...	...	632½	...	...	...	...	...	...	...	...
Mean		...	...	...	146	1,974	...	1.63	...	9.82	.198	...

SUMMARY—BASIN AND SEA-GOING TRIALS OF H. M. S. *SHELDRAKE.*

At the finish of the 1,000-mile trials, several experiments were made. The first one was on the forward boiler, when the time required to raise steam to 140 pounds pressure from cold water was taken; the temperature of the water at the start was 70 degrees, and steam was raised to 140 pounds pressure in 23 minutes. After that a stopping and starting test was made; the engines were going full speed and suddenly stopped. The front tube doors and uptake doors were immediately opened and the ash-pit doors closed; the steam gauge was then watched, and the pressure did not rise more than 5 pounds, neither did the safety valves lift.

The nxet test was made to ascertain how soon the operation of drawing a tube could be commenced after the fires had been pulled out of the furnace. No. 4 boiler was used for this purpose. This boiler was worked at full power; suddenly the fires were drawn and the water blown out; in 24 minutes after hauling the fires several caps were taken off ready for drawing tubes. Then a test was made to show how quickly a tube could be taken out of a boiler. Three tubes were drawn, one after the other; the first took 11 minutes, the second 10 minutes, and the third 9 minutes.

At the conclusion of these experiments, the *Sheldrake* had completed the whole of the Admiralty program and returned to Devonport, where a careful examination was made of the boilers, which were found to be in as good a condition everywhere as when they left the works.

Since the *Sheldrake* trials were carried out the following vessels in H. M. Navy have been fitted, or are about to be fitted, with these boilers:

Name.		Indicated horsepower.	Year.	No. of Babcock & Wilcox boilers.
Sloop	*Espiegle*	1,400	1899	4
Sloop	*Odin*	1,400	1900	4
Cruiser	*Challenger*	12,500	1900	12
Cruiser	*Hermes*	10,000	1901	12
Battleship	*Queen*	15,000	1901	15
Cruiser	*Cornwall*	22,000	1901	24
Battleship	*Dominion*	18,000	1902	16
Battleship	*Commonwealth*	18,000	1902	16
Battleship	*King Edward VII*	18,000	1902	10
Cruiser	*Argyll*	21,000	1902	16
Battleship	*Hindustan*	18,000	1902	18
Cruiser	*Black Prince*	23,500	1903	20
Cruiser	*Duke of Edinburgh*	23,500	1903	20
Battleship	*Britannia*	18,000	1904	18
Battleship	*Hibernia*	18,000	1904	18
Battleship	*Africa*	18,000	1904	18
Battleship	*Lord Nelson*	16,750	1904	15
Cruiser	*Minotaur*	27,000	1904	25
Battleship	*Dreadnought*	23,000	1905	18
Cruiser	*Indomitable*	41,000	1905	31
Battleship	*Bellerophon*	23,000	1906	18
Battleship	*Superb*	23,000	1906	18

The greater number of these vessels have already successfully passed their official trials, and some very interesting particulars of these are to hand from Messrs. Babcock & Wilcox, to whose courtesy we owe the accompanying illustrations.

In connection with the following details of trials it should be noted in every case that the fuel consumed is for all purposes, but the indicated horsepower is that of the main engines only.

H. M. S. *DREADNOUGHT.*

H. M. S. Dreadnought has eighteen Babcock & Wilcox boilers installed, having a total tube-heating surface of 55,530 square feet, and a total grate area of 1,599 square feet. Her very satisfactory trials are a matter of recent history. The evaporation at all trials worked out to between 10 to 10.2 pounds per pound of coal per hour, which, under the service conditions prevailing during the trials, must be regarded as an excellent result. During the course of the week's steaming it was decided to ascertain the maximum power which could be realized, and 28,000 indicated horsepower was developed ac-

cording to the torsion meter. This gives an overload, a power above the maximum provided for in the design, of 5,000 horsepower; the coal consumption with this overload equalled about 1.4 pounds per shaft horsepower per hour.

H. M. S. *KING EDWARD VII.*

H. M. S. King Edward VII has ten Babcock & Wilcox boilers, having a total tube-heating surface of 27,620 square feet, and a total grate area of 815.5 square feet; and six cylindrical boilers, having a total heating surface of 16,320 square feet, and a total grate area of 491 square feet.

RESULT OF OFFICIAL TRIALS, H. M. S. *KING EDWARD VII.*

.....	Low power.	Maximum continuous.	Full power.
Date of trial	Nov. 16th and 17th, 1904.	Nov. 18th and 19th, 1904.	Nov. 23d, 1904.
Duration of trial, hours	30.0	30.0	8.0
Number of boilers in use	6 Cylindrical	6 Cylindrical and 10 B. & W.	6 Cylindrical and 10 B. & W.
Heating surface, square feet	16,320.0	43,940.0	43,940.0
Grate area, square feet	491.0	1,306.5	1,306.5
Fuel, kind used	Welsh coal.	Welsh coal.	Welsh coal.
Steam, average observed gauge pressure, pounds per square inch	193.0	194.0	200.0
Draft, inches water pressure in stokehold	Nil.	Nil.	B. & W., 0.55 Cyl., 0.71
Indicated horsepower	3,760.0	12,844.0	18,138.0
Indicated horsepower per square foot of grate	7.66	9.83	13.88
Coal, total consumed per hour, pounds	9,889.0	25,174.0	39,359.0
Coal per indicated horsepower per hour, pounds	2.63	1.96	2.17
Coal per square foot of fire-bar surface per hour, pounds	20.1	19.2	30.0
Heating surface per indicated horsepower	4.34	3.42	2.42
Revolutions per minute	69.8	106.1	116.5

COMPARATIVE TRIALS TO OBTAIN RELATIVE COAL CONSUMPTION OF BABCOCK & WILCOX AND CYLINDRICAL BOILERS.

The official trials of this vessel, which is fitted with two-fifths cylindrical boilers and three-fifths Babcock & Wilcox boilers, not having been so satisfactory as those of similar vessels fitted entirely with Babcock & Wilcox boilers, a series of comparative trials were ordered to be carried out, using alternatively cylindrical boilers alone and Babcock & Wilcox boilers alone.

Before this was commenced, however, numerous alterations were made to the cylindrical boilers with the object, if

possible, of improving their performance, and they were fitted with retarders and generally in accordance with the most modern practice in all respects.

The conditions were that four trials were to be made, each of eight hours' duration. On two of these trials all the cylindrical boilers were to be used, and on the other two, a sufficient number of Babcock & Wilcox boilers were to be used to give the same proportion of the total power as the cylindrical boilers. The total power of the ship is 18,000 indicated horsepower. The cylindrical boilers were designed to give two-fifths of this power—namely, 7,200 indicated horsepower; and the proportion of the Babcock & Wilcox boilers used on the trials, was to be capable of giving the same proportion of the total power—viz: two-fifths—7,200 indicated horsepower.

On the first pair of trials each set of boilers was to develop 3,600 indicated horsepower.

COAL PER INDICATED HORSEPOWER.

On the second pair of trials the cylindrical boilers were to develop the maximum indicated horsepower of which they were capable, and then the Babcock & Wilcox boilers were to burn the same aggregate quantity of coal in order to ascertain what power could be obtained from them.

The results conclusively proved the superiority of the Babcock & Wilcox boilers, and were as follows:

.....	One-fifth power trial.		Full power trial.	
	Cylindrical.	Babcock & Wilcox.	Cylindrical.	Babcock & Wilcox.
Indicated horsepower.....	3,634.0	3,759.0	6,686.0	7,510.0
Coal per indicated horsepower.	1.8	1.74	1.88	1.67

It should be noted that the cylindrical boilers failed to obtain their designed power—viz: 7,200; whereas the Babcock & Wilcox boilers easily obtained it, and they could have developed a much larger power if required, their limitation being the total quantity of coal, which was not permitted to exceed that burnt in the cylindrical boilers.

.....	Low power.	Maximum continuous.	Full power.
Date of trial	Oct. 3d and 4th, 1906.	Oct. 6th and 7th, 1906.	Oct. 9th, 1906.
Duration of trial, hours	30.0	30.0	8.0
Number of boilers in use	9	18	18
Heating surface, square feet	27,765.0	55,530.0	55,530.0
Grate area, square feet	799½	1,599.0	1,599 0
Fuel, kind used	Welsh coal.	Welsh coal.	Welsh coal.
Steam, average observed gauge pressure, pounds per square inch	220.0	230.0	241.0
Draft, inches water pressure in stoke-hold	0.56	0.9	1.2
Indicated horsepower	5,013.0	16,930.0	24,712.0
Indicated horsepower per square foot of grate	6.27	10.58	15.45
Coal, total consumed per hour, pounds	12,955.0	28,815.0	37,315.0
Coal per indicated horsepower per hour, pounds	2.58	1.7	1.51
Coal per square foot of fire-bar surface per hour, pounds	16.2	18.02	23.33
Heating surface per indicated horsepower.	5.54	3.28	2.25

OFFICIAL TRIALS OF H. M. S. *DREADNOUGHT.*

The first-class battleship *Britannia,* of 18,000 indicated horsepower, is installed with 18 Babcock & Wilcox boilers, having a total tube heating surface of 40,020 square feet, and a total grate area of 1,250 square feet, and three cylindrical boilers, having a total heating surface of 8,100 square feet, and a total grate area of 247 square feet. Six of the 18 Babcock & Wilcox boilers are fitted with their patent superheaters.

THE NAVY REPAIRS QUESTION.

In the last Navy debate in Parliament Mr. Balfour pointed out that, so far as he knew, the whole disquiet of the public mind regarding the position of the Navy related to repairs, and in writing on this debate in a recent issue we promised to investigate this question. There is, as Mr. Balfour said, some disquiet in the public mind, and it is possible for the superficial critic to make some sort of a case against the Admiralty because of the number of ships awaiting repairs, each with a more or less important list of defects. But a closer examination of the whole situation, and an analysis of the comparative fighting efficiency of the ships on the list, reduces to vanishing point what is referred to by some writers as a serious situation.

Most of the ships, indeed, belong to the second line, and many of the defects do not concern the fighting machinery. For this and other reasons the Admiralty policy, while ensuring permanent economy, does not affect the preparedness of the Fleet to maintain our supremacy at sea.

The attention devoted to the repairs question is indeed an evidence of that public interest in the Fleet which is an important factor in the maintenance of its efficiency; the outcry is itself proof of this general efficiency. Under the conditions antecedent to 1903 a ship, after she had been in commission three years, and in some cases four years, returned to the dockyard, to be passed into the Dockyard Reserve. If it was not convenient to undertake the refitting or repairs necessary, she was quietly forgotten. Even when refitted and passed into the Fleet Reserve, she remained there without anyone being directly responsible for her maintenance in a satisfactory condition. A company of officers and men, it is true, boarded her from time to time; but as they were seldom called upon to go to sea with the ship, they had little direct interest in the ship or her machinery. The consequence of this inattention, if not neglect, was that when the vessel went on maneuvers with a more or less scratch screw, her performances were unavoidably unsatisfactory. Under present conditions a ship is expected, after she enters the Fleet new, to be in active service practically continuously until she becomes obsolescent. She is commissioned for two years. At the end of this time, if no extensive repairs are required, she is re-commissioned. If, however, changes are necessary, either as the result of wear and tear, or of current improvements in mechanical appliances, the vessel is passed into the Home Fleet, there to await a convenient opportunity for undertaking such moderate refit as is necessary. If she goes to the Nore Division, she has a full crew; if to the other divisions, a nucleus crew. If the refit required is extensive, the vessel is placed in dockyard hands and a temporary crew is appointed for care and maintenance purposes and to assist the dockyard in the repair work. The consequence is that at no time in the history of the ship is she with-

out a staff who is responsible for her efficiency. Thus, when repairs are called for, and the non-fulfillment of them, or delay in undertaking them, militates against efficiency, the officers naturally make it their business to enlist official and, in some cases, public influence to ensure that the repairs will be carried out. In the past there was the possibility of neglect without disclosure. Today there may be neglect, but the responsibility for non-efficiency comes home so directly that there must be disclosure. This is all for the good of the Service. Indeed, it is the duty of the commander-in-chief of each fleet to report periodically direct to the Board of Admiralty as to the state of his ships, and immediately in the event of any defect being discovered; so that here, again, there is advantage in having so many ships in commission, even if the crew be of nucleus strength.

Such disclosure of delay may be an explanation of the increased attention directed to the subject, as the extent of delay does not justify the outcry: the delay is not nearly so serious as in the past. It is difficult to measure the repair requirements of the Fleet. In industrial concerns, and in merchant fleets, experience has established a certain rate per cent. per annum for repairs; but there are very few concerns where this could be justified as accurate year by year; it is rather an arbitrary figure accepted for accounting. The case of the Fleet is still more difficult. The repairs of the Fleet depend on many factors—on the amount of work done by the ships and on the number of accidents partly consequent upon officers taking risks in maneuvers which would be inevitably forced upon them in war. There is also the question of the advance in invention, which may make small or extensive changes imperative. There is, for instance, the present case of supplying cold air for the effective ventilation of magazines, the need of which was established by research in the deterioration of explosives under high temperature. Equally operative is the age of the ships. The scrapping of several obsolescent ships a year or two ago necessarily reduced the repairs bill to a large extent. Finally, there is the amount of work undertaken by the ship's

company; and in this respect great advances, as will presently be explained, have been made in British warships. But notwithstanding this, from £1,500,000 to £2,000,000 are spent per annum; during the past five years £11,000,000 have been spent on repair work.

When, under the present régime, a scheme was elaborated in order to ensure efficiency in every department of the naval service, the organization equipment, and general system of repair work at the dockyards were included, and it was decided to discontinue the practice of giving out repair work to contract. At the same time it was considered desirable to reduce to the minimum the amount of new constructional work undertaken at the dockyards. As it reduced the probabilities of extensive refits the scrapping of a large number of old ships involved a reduction in the number of workmen employed at the dockyards. This was commendable, as experience had shown that there was lack of economy, because the number of workers could not be varied to suit the great fluctuations from time to time in the amount of repair work to be done. It was not possible to vary the number of men employed in the dockyards as in the Clyde district, for instance, where the opportunities for re-engagement in another establishment are so great that the men discharged seldom experience hardship. Under the old conditions the whole of the squadron returned at one time for refit, and there was a glut of work; at other times there was a paucity. The machinery of a ship was invariably opened up. If the defects discovered were extensive, and there was otherwise plenty of work, the ship was passed to the reserve, to be repaired at a more convenient season. Cylinders, bearings, &c., were as often as not left open, so that there was serious deterioration. Moreover, when there was little work to do, there was a tendency to create work, and machinery might be opened up in order to discover whether a job were possible, without any apparent reason, and thus money was wasted without the efficiency of the Service being improved, and simply to obviate the discharge of men. There is now no opening up of machinery and boilers by the dock-

yard, unless the engineer of the ship has reason to believe that such is desirable.

Instead of the whole of a squadron returning simultaneously for refit and repair, not more than three battleships and one armored cruiser can be detached from each fleet at any time for defects to be made good. Here we have a very important advantage in maintaining the preparedness of the Fleet for action. By this system there cannot be repeated the serious condition which prevailed at the time of the Dogger Bank incident, when the whole of the vessels of the cruiser squadron were undergoing repairs.

The number of men employed has been reduced to the normal staff considered suitable to meet the normal volume of repair and new constructional work. Great fluctuations are avoided without abnormal delay in refit work, undue creation of defects, or prolongation of work. If the number of workers is sufficiently large for well-organized repair work there is a certainty of a better efficiency. The whole question resolves itself into one of determining the urgency of repairs from the point of view of the efficiency of the Fleet and its preparedness for action. In this the Admiralty are only following private practice, which is based on sound commercial experience. It is true, of course, that the Navy must be maintained in efficiency, apart altogether from financial considerations; but that does not preclude the authorities from combining economy with efficiency. This latter term, too, is relative. If we were weak in any of the types of warships in comparison with probable opponents, it would be necessary to ensure that every vessel was at all times available for service; but the view is now accepted that our Fleet is at least numerically superior to the two-Power standard. It therefore becomes in part a question as to the relative number of vessels on the repair list in our own Navy and that of foreign Powers. We are satisfied that in this respect we are in an advantageous position under the present system. Consequently there must be approval of the principle now adopted of considering the importance of the defects of one ship against another, and of weighing the value

of each ship in maintaining the composition of our sea-going fleets. Thus the *Dreadnought* or the *King Edward VII* must have preference to a vessel of the *Royal Sovereign, Majestic* or *Canopus* classes. The same applies to cruisers. Again, it is incumbent upon the authorities to give preference to such repairs in any one ship as affect the fighting power of that ship. Thus it would be more important to replace the controlling valve of a hydraulic gun mounting than to repair the mechanism of an ash hoist. Both are important, but in very different degree. The urgency with which the work should be undertaken must vary in direct ratio to the influence which the defect has upon the fighting efficiency of the ship or of the Fleet.

Another consideration is that by concentrating, as far as necessary, attention upon the repairing of ships of high fighting efficiency, the preparedness of the Fleet is maintained at a higher standard, and the training of the officers and men made more continuous. When the active-service ratings are not sufficient to completely man all the vessels in the Fleet, it is better that they should man all the ships that are immediately ready to fight, leaving the remainder partially empty, for the dockyards to complete at high pressure when the contingency arises. The ships in reserve awaiting repairs may be manned by less efficient men from the Fleet or the Royal Naval Reserve. This provides a reserve of ships, which, in the event of war, would very quickly be brought to a state of efficiency by an emergency increase in the staff of the dockyards.

There seems no present reason for maintaining the dockyard staff at an abnormally high standard when there is not continuous work for them. If experience proves that under the present conditions in the dockyards it is not possible to obviate a permanent delay in the repair of the ships, then it will be easy to increase the numbers; but it has been demonstrated in the past that it is easier to engage than to get rid of men in the dockyards, and that surplus labor, in view of the absence of continuity in the volume of repairs, militates seriously against economy. That the staff is adequate has

been proved by the great increase in rapidity with which repair work is now carried out. Formerly a thorough refit for a battleship occupied anything from twenty to forty months, whereas now, without any overtime, six months suffice. Dilatory work is always expensive, and thus the rapid completion of repair work not only ensures the return of the unit to the fighting line in a much shorter period of time, but also makes for economy.

An important factor in the problem has reference to the amount of work done on board ship for the maintenance of machinery in a thorough state of repair. Now that each ship is continuously in commission, instead of being passed into the Fleet or Dockyard Reserves, officers naturally gain credit for the maintenance of their ship in a satisfactory condition, and are, we hope, commended for undertaking important work for the repair of their machinery. Every ship of a size larger than a destroyer has now a very extensive workshop. In a battleship, for instance, there are fitted power-driven horizontal boring machines, screw-cutting lathes, ranging up to fairly respectable capacity, a side-planing machine, various drilling machines, punching and shearing machines, tool-sharpening, twist-drill grinding, and other tools. There is a respectable sized forge, with furnaces and appliances for producing small castings. Some of the engineer officers have done very good work to meet emergencies. In one case that we know of a brass casting of 116 pounds weight was made with the appliances on board. The ships carry rough castings and unmanufactured articles, the use of which for renewal of parts can be anticipated. On occasion the engine-room artificer ratings are relieved of watch-keeping duties for this repair work. From several engineer officers we have heard of the enthusiasm of the artificers in this return to the work of their earlier years; they have often continued to work when they were supposed to be off watch. In every complement, too, there are usually found representatives of almost every trade associated with the manufacture of machinery, and the result is a spirit of emulation between men of each ship and between ships, which is most conducive to the efficiency of the Service.

Repair ships are now attached to the principal fleets, and these are to all intents and purposes floating factories, having a pattern shop, joiner's shop, foundry, smithy, boiler shop, coppersmith's and plumber's shops, and a machine and fitting shop for engines, boiler and electrical work. The forge and plate furnaces are usually run by motor fans. There are the usual slabs for bending plates, angles and bars. The foundries can undertake pretty heavy castings, and are fitted with electric cranes. The coppersmith's shop has the usual forges, slabs, and bending machines, &c. The carpenter's shop has circular and other machine saws and tools. As a consequence the ships do not require to come back to the dockyard so frequently for repairs.

It will be realized that the repair work has been thoroughly systematized as an important part of naval re-organization, and will, as in other departments, be conducive to the improvement of naval efficiency. We are convinced that the scheme will eventually effect a great economy without militating in the slightest degree against preparedness for war.—"Engineering."

"BACK FLASH" FROM MODERN SMOKLESS POWDER.

The recent terrible disaster on board of the Japanese battleship *Kashima,* in which over forty officers and men were killed and injured owing to charges of modified cordite, while waiting to be loaded into a 10-inch gun, being ignited by the "back-flash" from the previous charge just fired, once more directs serious attention to an ever-present danger which threatens our own Navy. Both nations use practically the same pattern of gun and absolutely identically the same "powder"—or, to be more precise, the same propellant—namely, modified cordite. The tale of these accidents is mounting up; the United States has to deplore the loss of many of her sailors from exactly the same cause, though the propellant used in the United States Navy is of a somewhat different type.

Briefly narrated, what happens is this: When a charge of

modern smokeless propellant is fired, the shot issues from the muzzle, but inside the bore of the gun there is left a large quantity of various mixed gases at a high temperature. Owing to their nature, however, they themselves are not sufficiently rich in oxygen to cause a visible flame. When the breech is opened there is an excess of fresh oxygen from the atmosphere, and the hot gases then burst into flame. If there is a draft down the bore of the gun from the muzzle to the breech, this flame is blown backwards a considerable distance to the rear of the gun, scorching and burning anything in its way. Should a charge of fresh propellant be ready behind the gun, waiting to be loaded after the shot is rammed home, it is not difficult to understand how there is certainty of terrible devastation, accentuated as it is by the small enclosed space in a turret, barbette or casemate, and the further possible danger of the flame spreading to charges in the ammunition hoists and thence down to the magazines.

As the velocity of the projectile is increased in each succeeding type of gun, the charges of propellant rendered necessary are greater, and thus the danger from back-flash is intensified, owing to the larger quantity of residual gases being left in the bore.

It was appreciated some time ago in our Navy that precautions must be taken to guard against such accidents as these, and the method since adopted is to expel the residual gases by an air-blast before the breech is opened; but this is at best a makeshift, and must unduly delay the service of the gun. Moreover, in the hurry of an action, sufficient time may not be given to the operation; nor if the ship is steaming to windward, or the guns pointing to windward, can it be safely depended upon to clear out all the gases.

The Germans seem to have solved the problem in a much more scientific and satisfactory manner by adopting for their Navy in 1906 a propellant powder which gives no back-flash, even in their largest guns. This has replaced in their Navy the modified cordite which is used at present in the British Navy. This final rejection, owing to the serious back-flash

defect, in favor of the flameless powder, known by the designation C/06, is a consequence of searching trials, extending over four years since 1902, the " /06" denoting the year in which it was adopted.

The exact composition of this powder is kept secret, but it consists of a mixture of nitro-glycerine and nitro-cellulose and vaseline, in which, so far, it closely resembles the composition of our modified cordite; but there is, in addition, a small quantity of some chemical which has the effect of preventing back-flash at the breech, and in smaller ordnance no flame is apparent at the muzzle. This latter is a considerable advantage during night firing, as the gun layer of the quick-firing weapon is not blinded by the intense brilliancy of the flame usual with our cordite, whereby the speed of firing is reduced and the accuracy of shooting impaired. It appears that the new German powder gives excellent shooting results.

So far as keeping qualities are concerned, the German powder is even better than our own cordite. This also is an important factor in view of the cordite explosions which have taken place from time to time in the magazines of British war vessels, fortunately, so far, without loss of life or of a ship; but similar good fortune may not continue. The report on these explosions has been kept extremely quiet, so that it is difficult to speak more fully of them; but on one occasion it was mere luck that a battleship was not sunk with all hands. Owing to this immunity from disaster, the public have not had their faith in the safety of ships in our fleet shaken in the same way as France quite lately was rudely awakened from her confidence in the excellence of the propellant powder used in her Navy by the terrible disaster on the *Jena*. Apart, however, from the question of instability of a powder, which can be met to a large extent by the installation of refrigerating machinery for the magazines, there remains the equally important question of doing away with back-flash. So far no mechanical means have been devised to entirely obviate the danger therefrom, and recourse must therefore be had to the chemist's art.

Germany has always been renowned for her progress in

chemistry, and in no branch has she shown herself more progressive than in matters connected with explosives. There is no single authenticated case of which we know of where an explosion has occurred on a German vessel of war, and the reason for this seems to lie in an intelligent anticipation of possible sources of danger and in the steps taken to prevent them. For example, as soon as nitro-smokeless powders replaced the smoke powders it was at once realized that the dictum of former days "keep your powder dry" must be changed to "keep your powder cool." All ships in the German Navy were fitted with apparatus to keep the magazines cool, but our Admiralty only seem to have appreciated the importance of this during the past year, when the matter was forced on the attention of the Government by the accident which occurred on H.M.S. *Fox,* though several years previously a similar accident had occurred on H.M.S. *Revenge.* In both cases, by a simple chance, the loss of the ships and their crews, fortunately, did not occur.

Now, again, it would appear that we shall have to follow the lead given by Germany and adopt a powder which will obviate the danger of back-flash in a proper and scientific manner. The grave fault in our administration appears to be that it is apparently no particular naval official's duty to watch the question of explosives and the progress made therein abroad. It is true that committees almost innumerable sit and report, but definite and prompt action does not appear to be taken upon the recommendations, or it is so leisurely that by the time the suggestions of the committee are put into effect they are practically out of date.

Committees themselves are not the most efficient methods of ensuring prompt reform. The members have frequently other duties, and are, moreover, not always suited by training and experience to properly consider the terms of reference. The result is that many of the earlier sittings are wasted in the elementary education of members in the subject under consideration. The sittings, too, are seldom, if ever, continuous; not more than an hour or so each week being devoted to effect-

ive investigation, and the period is still further broken up by holidays. Meanwhile much of the experimental work for the Committees is unnecessarily delayed. Finally, when the recommendations are made there is too often difference of opinion due to a lack of full knowledge of the subject. The crux of the matter is therefore the concentration of responsibility by the appointment of an officer solely for explosives, whose duty it would be to acquire knowledge of foreign progress, maintain continuity in the records of British experience, and analyze the results of all operations in the Navy and in the private works devoted to research in explosive compounds. We are satisfied that with a fuller coordination between experimental work in private factories and naval battle practice, and with a firmer determination to overcome difficulties, our Admiralty would in this, as in other departments of naval activity, maintain our prestige.—"Engineering."

BATTLESHIP STRENGTH AND RELATIVE VALUE.

With October comes the intersessional political oratory, and this year we are promised a more than usually vigorous campaign. Two questions of interest to all connected with industry will be brought to the front: The one connected with trades-union organization and legislation, the other associated with naval strength and Admiralty administration. The former topic can only be considered when the demands of the labor socialist have been formulated. As regards the latter subject, the facts are indisputably in favor of the Admiralty policy during the past three years or so. We shall, as ever, be confronted with lists which state that Great Britain possesses so many battleships, that France, Germany, the United States of America, &c., possess so many; and starting from this hypothesis, articles will be written to prove that either now, or at some date in the comparatively near future, the first-named Power will be overtaken and passed, and that the sea supremacy, which none deny to her in the present, will then be dangerously jeopardized. But the writers and speakers who merely count

units—units which are described as battleships—do not always analyze, and lay before the public which they address, the data concerning the vessels mentioned in these lists.

If a comparison is to be of any value, it must take into account not only the number of ships, but also their fighting capacity. As an example let us see in what way the Channel Fleet, under the command of Lord Charles Beresford, will be constituted in the immediate future. It will consist of eight ships of the *King Edward VII* type and six of the *Formidable* type, fourteen battleships, forming the most powerful and also the most homogeneous force, which is an all-important matter, in the whole world. Of the first type mentioned, the oldest was launched in 1903, the youngest—the *Hibernia*—in 1905. These eight ships each carry four 12-inch, four 9.2-inch and ten 6-inch guns, as well as small guns and five submerged torpedo tubes; their armor is 9-inch Krupp amidships, tapering to 6-inch and 2-inch; their horsepower is 18,000, and their speed nearly 19 knots, their tonnage being 16,350 tons. The *Formidables* are older, November, 1898, being the date of the oldest of the class, and their size is smaller, being 15,000 tons, carrying four 12-inch, twelve 6-inch, besides small guns and four submerged tubes; the horsepower is 15,000, and speed 18 knots. These details are given in order that comparisons may be instituted between this, the principal British Fleet, and the ships of foreign Powers; showing the relative strength of the units which go to make up their fleets, as well as our own. The Home Fleet, which contains the *Dreadnought,* and the Atlantic Fleet are purposely left on one side in order to avoid confusion.

The Navy of the United States of America has, of late years, outstripped that of France and taken second place, next to our own. Undoubtedly, however, the *Dreadnought* took the Americans by surprise, and since her trials and the demonstration that this new departure was a success, they have been attempting to make up their leeway. If we go back to 1898, the date of the oldest of the *Formidables,* we shall find that the United States built in that year the *Kearsarge* and *Kentucky,*

of 11,500 tons, carrying four 13-inch, four 8-inch and fourteen 6-inch guns, with a 16½-inch Harvey-nickel armor belt amidships; the horsepower is 10,500, and speed 16 knots. In the same year were built the practically identical ships *Alabama, Illinois* and *Wisconsin*—as far as tonnage was concerned—but the armament of which differs considerably from that of the *Kearsarge* and *Kentucky,* as, instead of four 8-inch, they carry fourteen 6-inch guns. It will be observed that the *Formidables* are superior in nearly every respect, and have the enormous advantage of homogeneity. In 1901 the United States built the *Maine* class, of three ships, of a tonnage of 12,500, carrying as their main armament four 12-inch and sixteen 6-inch guns; horsepower, 16,000, and speed, 18 knots. In 1904 came the *New Jersey* class, of five ships, of 14,948 tons, with four 12-inch, eight 8-inch and twelve 6-inch guns; horsepower, 19,000, and speed, 19 knots. In the same year came the *Louisiana* and *Connecticut,* of 16,000 tons, with practically the same armament; and in 1905 came a reversion to a smaller type—the *Idaho* and *Mississippi,* of 13,000 tons, with eight 7-inch guns, instead of twelve 6-inch. The *Kansas* class, of 16,000 tons, also date from 1905, and carry the same guns, except that they have twelve, instead of eight 7-inch guns, and that with a horsepower of 16,500 their speed is 18 knots. In 1906 the first of the *South Carolina* class was laid down; and in this ship, and her sister, the *Michigan,* comes a foreshadowing of the future, as with a displacement of 16,000 tons their armament is eight 12-inch guns, the speed and horsepower remaining the same as in the case of the *Kansas* class. But the *Dreadnought* had seized upon the American imagination, ever prone to indulge itself, both by sea and land, with the biggest thing going, and two ships, forming the *Delaware* class, are to be laid down during the present year; their displacement is reckoned at 20,000 tons, and their armament will be ten 12-inch guns. In considering the available battleships of the United States we shall see that, supposing them all to be in commission and ready for sea—which they are not—they could oppose 21 to the 14 of our Channel Fleet (which are ready and in commission), that five

of these are vessels of 11,500 tons, and that homogeneity would be far to seek. Of personnel it may be stated in passing that the United States Navy is short some thousands of men; and although the officers responsible say that this deficit could easily be filled, it cannot be considered that the Navy is an efficient service until this is done; and, in spite of official optimism, the supply of free-born white Americans who seek to gain a living on the sea is by no means on the increase.

The importance of the French Navy has been always reckoned as one of the most potent factors in keeping the balance of European armaments; but it cannot be said that of late years the great Republic has kept up her ancient fame in this respect. This is no place to deal with the political aspect of affairs, but the man who has followed with intelligent interest the doings in the French Navy, and the French arsenals, for the last five or six years, would be blind and deaf to patent facts did he not recognize the enormous mischief wrought, both in the disciplined service and in the great industrial undertakings with which its upkeep is maintained, at the hands of a socialistic Minister of Marine. Dearly is M. Gaston Thomson paying for the policy of his predecessor M. Camille Pelletan, the Socialist, and dearly are the country and the Government paying in bad discipline and slackness among its employés, as well as in the more vulgar and tangible asset of hard cash. Thus it is that in relation to the French fleet of the present day one finds some difficulty when it comes to comparison of battleship strength, as no battleships were laid down for more than three years, and the last of the 1900 program has only just left Saint-Nazaire for Brest, to begin her trials. Not only was incalculable mischief done to discipline and the morale of the fleet and dockyards—witness the strike of the Inscrits Maritimes at Toulon three summers ago, and the antics of that most militant union, the "Syndicat Rouge," in the dockyards; but also the Minister considered that, in addition to anything else, it behooved him personally to advise as to the class of vessel upon which the naval credits voted were to be spent. M. Pelletan decided against battle-

ships, and perpetuated the error of Gabriel Charmes and the "jeune marine" of the early 'eighties, who declared that the day of the "capital ship" had come to an end as soon as the automobile torpedo became a practical and accomplished fact. Gabriel Charmes and his school pinned their faith upon torpedo boats of great speed, but the sea soon demonstrated that such craft were of small value in heavy weather. The submarine, in like manner, became the obsession of M. Pelletan, and on the submarine and the "petite marine," or small craft, has all the money been spent of late years. At last, however, saner counsels have prevailed, and France has started to make up all the leeway that she has lost.

It is impossible in the present day to consider that the *Charlemagne* class, of three ships which date from 1895, are of very much fighting value. They are, however, still on the active strength of the Navy; they are 11,200 tons, and carry four 12-inch, and ten 5.5-inch guns. The *Iéna,* dating 1898, was, as all the world knows, blown up at Toulon in March of the present year. The *Suffren,* 1899, is 12,750 tons, and carries four 12-inch and ten 6.4-inch guns; horsepower, 16,200; speed, 18 knots. The list closes with the *Republique* (1902) and *Patrie* (1903), of 14,865 tons, carrying four 12-inch and eighteen 6.4-inch guns; and the *Liberté* class, of four ships, of 14,900 tons, with four 12-inch and ten 7.6-inch guns; horsepower, 18,000; speed, 18 knots. But "mastodonte," as the French aptly call the craze for big ships, has seized also upon the Republic. At last common sense has regained her sway, and with it recognition of the fact that to be strong on the sea you must possess ships capable of lying in the line-of-battle. Therefore there have been laid down six ships—the *Danton* class, of 18,400 tons, to carry four 12-inch and twelve 9.4-inch guns. Further than this, it has been recognized that the worst of all economies is to keep ships too long upon the stocks, and in consequence, instead of seven years, it is designed that the *Danton* and her five sister ships shall pass into the active service four years from the date upon which their keel plates were laid.

Their fleet, both war and mercantile, is among those things

of which the Germans are the most proud—and with reason. The Kaiser on a memorable occasion declared that "the future of Germany is on the sea," and to do him justice, he has never ceased, in season and out of season, to do his best to ensure that that future shall be prosperous. Germany, like every other nation, piles ship upon ship and gun upon gun, protesting loudly all the while that nothing is further from her thoughts than war; this, of course, is all part of the game, and nobody minds, because we all do and say the same thing. But it may be permissible to doubt whether Germany does not view the development of "mastodonte" in warships with greater dismay than any other nation. The Fatherland was really getting along very nicely in the naval way until the arrival of the *Dreadnought* put all her plans astray. There is one thing that neither King, Kaiser nor Republican President can alter, and that is the physical configuration of the land in which they dwell, and Nature has ensconced Germany behind one of the most intricate and tortuous labyrinths of sandbanks which exists in the world. Very useful are such natural defences against a potential maritime enemy, but when a nation wishes to develop into a great maritime Power, they embarrass it almost as much as they would do its foe. As at present constituted, the Germany Navy possesses no battleship of over 13,200 tons, and has afloat no gun of a greater caliber than 11-inch. There are five of the *Braunschweig* class, dating 1902-3, and five of the *Deutschland* class, dating 1904-6, of this tonnage, carrying four 11-inch and fourteen 6.7-inch guns; horsepower, 16,000; speed, 18 knots. There are also five *Wittelsbachs,* of 11,830 tons, and five of the *Kaiser* class, of 11,150 tons, carrying four 9.4-inch and eighteen 6-inch guns, 15,000 and 14,000 horsepower, and speed 18 knots. The *Brandenberg* class, of four ships, date from 1891, and are of 10,060 tons,

If we compare these ships, of which the German high sea fleet is composed, with the Channel Fleet alone, we need not, at all events, fear the comparison. But Germany has further ambitions, and the "mastodons" which she has projected are

the *Sachsen, Baiern, Baden* and *Wurtemburg*. They are to have a tonnage, it is reported, of 17,710 tons, possibly 19,000 tons—but apparently nothing definite has been settled—and are to carry sixteen 11-inch guns.

The most modern battleships possessed by Japan are the *Kashima* and *Katori,* dating 1905; they are 16,400 tons, carry four 12-inch, four 10-inch and twelve 6-inch guns, have a 9-inch Krupp belt amidships, horsepower 17,000, and speed 18.5 knots. Our allies, however, are perhaps, further advanced than any other nation except ourselves along the road of big-ship building, as the *Satsuma,* of 18,800 tons, is already in the water, and her sister ship is advancing towards completion. There are also one ship, as yet unnamed, building, and one projected of 20,750 tons.

Of Russia there is but small need to speak at present, as her Navy is in an embryonic stage. Even when it is completed, it will not rank very high in the estimation of the world unless its methods of training are vastly different from those which obtained before the Russo-Jananese war.

Italy projects to lay down three or four ships of 16,000 tons; but those remarkably able men, the Italian constructors, have always held that for their country moderate tonnage would suffice, and today 16,000 tons represent moderation.

Austria is more modest still, as 14,500 tons is the limit, as far as is known, to which she will go at present in big-ship construction. There are other countries also which possess warships; but, as far as battleships are concerned, they may at present be left on one side.

The conclusion of the whole matter would appear to be that, for the present, England is in a satisfactory position. But the pace is being forced all along the line, and our rivals are striving to overtake and to surpass us. If this fact is kept in mind by our politicians, we need not fear for our supremacy, for we can build faster than any other nation, and certainly quite as well. The danger is that we may presume on our abilities, and defer our preparations too long. Fortunately, all parties are agreed that the British Navy must always remain supreme on the seas.—"Engineering."

POWER ESTIMATING FOR TURBINE STEAMERS.

The fact that the marine turbine cannot be "indicated" in the ordinary sense, and that nearly three-quarters of a million turbine "equivalent horsepower" have been and are being built for merchant vessels alone, makes us ask what is going to become of the Admiralty coefficient? It is unlikely to die out for many years, but for long enough it has been notoriously untrustworthy, especially for fast vessels, and now that "indicated" horsepower bids fair to become a thing of the past for every steamer of 19 knots and upwards, its use may become less frequent, and will certainly entail more caution than ever. Or will it gradually commence a fresh spell of useful existence in a more modern form?

There is one great feature about the adoption of marine turbines, which is that they have made naval architects and marine engineers think on new lines. Empirical methods of design, and a discreet copying of the last similar job, sufficed for the vast majority of work done for the mercantile marine when the comparatively slow-running engine and propeller were adopted. In their case the margins were large in all directions; the propeller was generally nowhere near its breaking-down point, the link motion enabled a greater mean pressure to be used if the required power was more than the estimate, analyses of component losses were almost invariably ignored, and with one or two exceptions, shipbuilders were content to trust the indicator, and base everything thereon. Consequently vital variables remained undisturbed for want of investigation that the turbine has since enforced, and lest anyone should be unduly assisted by the work of his predecessors, providence seems to have carefully arranged that this casual satisfaction with indicated horsepower and speed should also extend to the stokehold, and that the coal consumption should be measured while the water evaporated—and hence boiler efficiency as well—should be left undetermined. Consequently there arose a term—pounds of coal per indicated horsepower hour—that, while based on good experience with innumerable vessels, was absolutely useless for scientific comparison of results. For the

moment we are not so much concerned with the boiler end of estimating as with the term indicated horsepower. For up-to-date fast steamers it is as useless as nominal horsepower, as Kirk's analysis or Rankin's augmented surface methods, excellent as they may have been in their day. Times change, and what we have to face now is the problem of foretelling the effective thrust needed to propel a hull of given dimensions at a given speed. The term that marine engineers will not only have to get into the way of using, but also of estimating, will be effective horsepower, and *not* indicated horsepower. Now, the horsepower actually required to propel a given hull has to overcome the (1) frictional resistance of the hull; (2) the wave-making resistance; (3) the eddy resistance. The first of these can be calculated with considerable accuracy, though it is customary, by taking a higher value for the coefficient of friction than that originally found by Froude, to include eddy resistance with it, owing to the impossibility of separately determining the resistance due to this cause. Wave-making resistance is hard to determine without a tank, though some empirical coefficients do exist that give a close approximation for certain types of vessel, and careful analytical estimates from similar ships help us to determine the percentage of the total that it represents. In the case of tanks, when the resistance of the naked model has been determined, the effective horsepower is first calculated and the probable indicated horsepower arrived at by means of propulsive coefficients made up as follows: About 5 per cent. is added to the E.H.P. of the naked hull for resistance due to appendages—rudder, bossing, shaft brackets, &c.—and the figure thus found is multiplied by about $\frac{1}{.85}$ for engine efficiency, and about $\frac{1}{.65}$ for propeller efficiency, and by a very small corrective for hull efficiency, that is, for the difference between wake gain and thrust augmentation due to the action of the screw. The net result is that the ratio of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ is about 50 per cent., but varies very con-

siderably. For most vessels the value is between 48 per cent. and 56 per cent. at full speed, but instances of greater and lesser values could easily be quoted. It is an interesting question as to how accurate tank horsepower is. Models of the same ship tried in different tanks give amazingly variable results, but with care and experience the resistance can be determined very closely.

Consider for the moment the analysis of power from the engine end. The indicated horsepower as given by the indicator is considerably in excess of that delivered to the propeller boss on account of friction in the moving parts. Until the genesis of the torsion meter about four years ago it was very difficult to say what this loss was, as the testing of large marine engines against a brake was prohibitively costly. It was generally assumed—largely on the basis of trials of land engines against brakes or generators—to vary from 10 to 16 per cent., and it came as a considerable surprise to many when the Vulcan Company proved the mechanical efficiency of the main engines of the *Kaiser Wilhelm II* to be as high as 94 per cent. in service. Even assuming the accuracy of tank trials, it was therefore impossible to determine propeller efficiency correctly owing to the uncertainty as to the power actually delivered to the screw. Now that this can be done with comparative accuracy, it is so far only attempted on turbine steamers, and we find ourselves in the position of gradually commencing to accumulate isolated facts on the subject, but without any means of comparing them with reciprocating practice, as torsion meters are very rarely applied to piston engines. When, therefore, it becomes necessary to estimate the power required for a turbine steamer, and consequently revolutions of shafts, weight of turbines, &c., there is no guide established yet by past practice enabling us to decide exactly what brake horsepower is required. If a tank is available, or rather if a store of tank data is at hand to estimate from, the matter is simplified at once. If not, the procedure is guesswork. An Admiralty coefficient for indicated horsepower is taken as if piston engines were to be used, and an equally approximate propulsive coefficient is

assumed in order to arrive at the E.H.P. Consequently we find some cases of large margins, and others cut very fine indeed.

With the improvement in steamship performance the value of the coefficient should be rising in ships of all types; but as far as Channel steamers are concerned, there is now no indicated horsepower, and hence no trustworthy coefficient. The solution seems to be to substitute E.H.P. for I.H.P. wherever it can be done. It applies equally to both types of engine, and obviates the inclusion of undetermined losses in important estimates. The known propulsive efficiencies can be used, but the absolute values will change. Instead of taking, say, 240 as a suitable Channel steamer coefficient whose propulsive efficiency is known to be 54 per cent., our value in future will be 444. Unless some step like this is taken, and a new series of values compiled for a most useful formula, we are afraid that for all fast steamers its use and value for power estimating will rapidly wane. The same arguments apply equally well to proportions of boilers, to fuel consumption or relative weights; but at the present time we are in a transitory state on this question, and it is by no means easy to change over from the old-established symbols.—"The Engineer."

THE FRENCH NAVAL ARMAMENT.

The program of naval construction for 1909 now under consideration by the Conseil Supérieur de la Marine promises to be an unusually important one. In view of the limited expenditure upon new vessels in the coming year, it was anticipated that the Minister of the Marine, by restricting his efforts to the six battleships of 18,350 tons now being built under last year's program, was preparing for a much bigger development in the early future. The necessity for clearing off arrears and carrying out reforms in the arsenals and shipyards appeared sufficient to justify the small number of units which is to be put on the stocks during 1908, but a still more important factor

in determining the Minister to postpone the final preparation of his program is the uncertainty that has existed concerning the armament that should be carried on future battleships. While submarines, submersibles and torpedo boats will continue to form an important part of the French Navy, it is obvious that so far as heavy units are concerned the Conseil Supérieur de la Marine intends to follow the lead of other nations by equipping the fleet with as many powerful battleships as possible. It is doubtful whether many, if any, armored cruisers will be included in the list of constructions for 1909, which seems likely to be almost entirely comprised of battleships of 20,000 to 21,000 tons. Fitted with turbine machinery, they are expected to have a speed of 20 knots. Before deciding upon the dimensions to be given to these new heavy units, the Conseil Supérieur de la Marine is now holding frequent sittings with a view to discussing the armament that will be carried. It appears that the Conseil has five proposals under consideration. The first is to equip the future battleship with 12 guns of 305 mm. bore; the second, to increase the number of these guns to 14; the third, to employ 16 guns of 274 mm. bore; the fourth, to have 20 guns of 240 mm. bore; and the fifth, to adopt a compromise with 8 guns of 305 mm. and 8 of 240 mm. bore. There is apparently a strong feeling in favor of the third of these proposals—that is to say, adopting a numerous armament of 274 mm. guns—because it is argued that this type of ordnance is capable of proving very effective at a distance of 7,000 m., which is regarded as the extreme fighting range. It is claimed that at a distance of 7,000 m. the gun of 274 mm. bore will pierce armor 330 mm. thick under an incidence of 20 degrees. The advantage of having sixteen guns of the same caliber is that a greater quantity of shells can be carried than would be the case if there were 305 mm. guns on board, while the number would permit of a concentrated firing which might prove very deadly at the maximum fighting range. All this depends, however, upon whether the future developments in the protection of battleships will not nullify the effect of shells from 274 mm. guns at the extreme fighting

range. As it is expected that the Conseil Supérieur de la Marine will very shortly come to a final decision upon this question of armament, it will be interesting to see whether it has more faith in the guns or in defensive armor. Whatever solution is come to, it is now an article of faith in naval artillery circles that the gun caliber on battleships should be as uniform as possible, whether there is a tendency to suppress guns of medium caliber and employ little or nothing between the heavy ordnance and the light quick-firing guns for resisting torpedo attacks.—"The Engineer."

NAVAL WORK ON THE CLYDE.

With the consignment to the water of the first-class cruiser *Inflexible* from the stocks of John Brown & Co., Limited, Clydebank, on June 26th, naval work on the Clyde, so far at least as the building berths are concerned, became a negative quantity. Little has happened since to change this condition of affairs; but a number of smaller miscellaneous commissions from Admiralty quarters have kept whetting the shipbuilders' appetite for "more." At the moment of writing also a statement is published, on what seems trustworthy authority, to the effect that a well-known shipbuilding and engineering firm on the upper reaches of the river whose building stocks are almost entirely denuded of work of any kind has just booked, amongst other important orders for merchant steamships, a contract to build and engine for the Japanese Government a battleship of about 18,000 tons displacement and of great beam. This requires stronger confirmation, but everything points to its being true in essentials, though details which are not here repeated may be wanting in accuracy.

The *Inflexible,* owing mainly to the discovery of some flaws in her stern-tube castings shortly before the time appointed for launching, and to the delay which the removal of the imperfect and the fitting of perfect castings caused, was some three months later in being sent off the stocks than her sister ship *Indomitable,* which the Fairfield Company

launched in March. The work on other parts of the Clydebank vessel, however, went on as usual, with the result that the cruiser when launched in June, almost three months after the time originally fixed, was in a correspondingly advanced state. Boilers and machinery have been put on board with despatch, and it is hoped that she will be ready for steam trials in six or seven months' time. In all probability the three sister ships—the *Indomitable*, whose fitting out at Fairfield basin is proceeding apace, and the *Invincible* on the Tyne—will all be put into commission during August, 1908.

The battleship *Agamemnon*, which her builders, William Beardmore & Co., Limited, Dalmuir, took down the Clyde about the end of July, after drydocking at Govan for hull cleaning, at the Tail of the Bank, took in coal, adjusted compasses, &c. On August 5th she was boarded by a navigation crew sent from Devonport, and on the 19th she went on a preliminary trial on the Firth. Next day she proceeded down Channel on a thirty-hours' trial at one-fifth power, and after returning to the anchorage it is intended that she shall proceed on a further thirty-hours' trial at seven-tenths power.

A small but interesting proportion of the new work recently placed with Clyde builders consists of Admiralty contracts. The Ardrossan Dry Dock and Shipbuilding Company, Limited, has secured an order from the Admiralty for three horse boats, and the Grangemouth and Greenock Dockyard Company has been entrusted by the Admiralty with the order for five vessels for transport purposes. The latter order will be executed by the company at its Grangemouth yard, and the vessels will be towed to Portsmouth. Some time ago this company contracted to build for the dockyard at Malta three caissons, and to equip them complete with hydraulic machinery. These have now been completed and duly tested in the presence of Admiralty and Dockyard officials. Tested under hydraulic pressure corresponding to the duty they will have to undergo when in position, each of the caissons was opened and closed in the space of 4½ minutes, or half a minute under con-

tract time. The dimensions of the caissons are 96 feet by 42 feet by 22 feet each, weighing 750 tons. They have lowering and lifting bridges on the "Kinniple" principle.

Although not yet in full swing, the new works established on the Clyde, at Scotstoun, by Yarrow & Co., Limited, builders of torpedo boats and other light and high-speed craft, may be said to have been set in motion. The keels of two torpedo boats for the Government have been laid, and other berths are being prepared. The shipyard has a river frontage of about 780 feet, and a breadth of about 710 feet, and the firm has the option of purchasing more ground to the east. Seven slips are being laid out, and the general equipment of the works should make the yard one of the most perfect of the kind in the world. The extensive shops of the engineering and boiler-making departments are being rapidly equipped with the most up-to-date machinery. A feature of the works will be the fitting-out basin, which is now nearing completion. It is 300 feet in length, by 80 feet in width, and it is being covered in such a way as to allow fitting-out work to proceed in all kinds of weather. When the boiler-making shops—to deal with the full demand for the Yarrow water-tube boilers—and the several other departments are fully set going, it is understood that fully 2,000 workmen will be employed.—"The Engineer."

ECONOMY TESTS OF A 7,500 KW. WESTINGHOUSE-PARSONS STEAM TURBINE.

By J. R. Bibbins, Jr., M. Am. Soc. M. E.

The data herein presented comprises the principal results obtained on Sept. 1, 1907, during an 8-hour economy test upon turbine No. 253, installed earlier in the year, at Waterside station No. 2, of the New York Edison Co., New York City. This test was conducted entirely by the New York Edison Co., under the direction of Mr. J. P. Sparrow, Chief Engineer. The various arrangements therefor were carried out in accordance with a mutual agreement between builder and operator,

entered into previous to the test, and the results were obtained by independent computation.

The turbine unit tested is a standard Westinghouse type having a maximum rated capacity of 11,250 kw., and was built to operate on 175 pounds steam pressure, 28 inches vacuum and 100 degrees superheat. Under these conditions the turbine unit was guaranteed to have a minimum steam consumption of 15.9 pounds per kw. hour at the generator terminals with a normal speed of 750 r.p.m. Incidentally, the electrical efficiency of the generator was guaranteed to be 97.8 per cent., exclusive of friction and windage, at a load corresponding to that sustained during the test. The results of the test detailed below show an economy about 7.5 per cent. better than the guarantee.

METHODS OF CONDUCTING THE TEST.

During the test period, No. 2 Waterside station sustained practically all of the 25-cycle load on the system. Of this the unit under test carried practically 70 per cent., the other turbine units in the station assuming the remainder. This load was maintained as constant as possible by remote control of the turbine governor by the switchboard operator. Between the first and the last hours of the test the maximum variation in load was held within 4 per cent. above and below mean. During the last hour, however, the load decreased somewhat. Previous to the test this turbine unit had been running on a load of 7,000 kw., which was increased to its test load ten minutes before the start.

Three-phase electrical load was measured by the two-watt-meter method, using two Weston indicating watt-meters of the standard laboratory type. These instruments were calibrated at the New York Electrical Testing Laboratories immediately before and after the test. Power factor was maintained substantially at unity, and all electrical readings were taken at one-minute intervals.

As a surface condenser was used in connection with this turbine unit, the water rate was determined by weighing the condensed steam delivered from the condenser hot well. This

condensation was weighed in a tank mounted upon platform scales, with a reservoir above large enough to hold the condensation accumulating between each weighing.

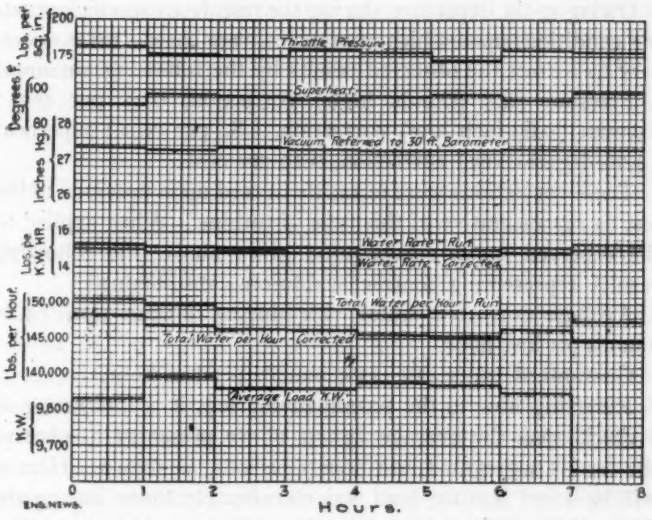
These weighings of 12,000 to 13,000 pounds each were made at intervals of 5 minutes. By a loop method of connecting the gland water supply, the necessity for correcting condensation by an amount equivalent to the weight of the gland water used is avoided.

As the circulating water is quite salt, any condenser leakage may immediately be detected by the salinity of the condensed steam, which should be pure distilled water. On this account condenser leakage was determined entirely by chemical analysis, employing the silver-nitrate test with a suitable color indicator. This method proved extremely sensitive and possessed a decided advantage over the ordinary method of weighing the leakage accumulating during a definite period when the condenser is idle and under full vacuum. As samples of circulating water and condensed steam could be taken at the same time, this method made it possible to discover any change in the rate of condenser leakage taking place during the test, while the method of weighing above described provides only an average result during the period.

In this condensing plant the delivery of the hot-well pump is automatically controlled by a float valve in the interior of the hot well. This maintains the water level therein at a practically constant point, and hence no correction had to be made for difference in level of water in the hot well before and after the test.

Steam pressures and temperatures were determined close to the turbine throttle. As usual, the degree of superheat was obtained by subtracting from the actual steam temperature the temperature of saturated steam at the corresponding pressure carried at the time. All gauges and thermometers were calibrated previous to the test at the U. S. Testing Bureau. It will be noted that both pressure and superheat were somewhat below the guarantee.

Vacuum was measured directly at the turbine exhaust by



PLOTTED LOG OF TESTS OF A 7,500-KW. WESTINGHOUSE-PARSONS STEAM TURBINE.

means of a mercury column with a barometer alongside for reducing to standard barometer—30 inches. This also obviated the necessity for temperature correction between the two mercury columns. During the test the vacuum was not maintained quite up to normal.

RESULTS OF TESTS.

The following data represents the results of the tests, calculated for the conditions as actually run, i. e., for instrumental errors only:

Duration of test, 9:30 A. M. to 5:30 P. M.

Average steam pressure at throttle, pounds, per square inch gauge, 177.5.

Average superheat at throttle, degrees F., 95.74.

Average vacuum (referred to 30 inch barom.) in Hg., 27.31.

Average load on generator, kw., 9,830.48.

Average steam consumption, as tested, pounds, per kw.-hour, 15.15.

Owing to the departure, during the test, from specific operating conditions upon which guarantees were based, it was necessary to correct the observed results by the following amounts:

Pressure (2.5 pounds high) correction, 0.25 per cent.; vacuum (0.69-inch low) correction, 1.84 per cent.; superheat (4.26 degrees low) correction, 0.29 per cent.

These corrections were mutually agreed upon previous to the test as representative of this type of turbine. When applied to the observed steam consumption given above, the following results, representing contract conditions, are obtained:

Average corrected water rate during 8-hour test, 14.85 pounds per kw.-hour.

Guaranteed water rate, 15.9 pounds per kw.-hour.

Referring now to the accompanying log, it is interesting as a check upon the average figures above presented to observe the results segregated into hourly periods, as shown. Here it will be noted that the load was considerably lower during the first and last hour than during the main part of the test. Neglecting, therefore, these two hours and considering only the six-hour period from 10:30 A. M. to 4:30 P. M., the results are as follows:

Average corrected water rate, 14.8 pounds per kw.-hour.

Equivalent water rate, 10.65 pounds per B.HP.-hour.

Equivalent water rate, 9.8 pounds per I.HP.-hour.

The two latter quantities are determined by applying conversion factors for generator efficiency and for internal losses.

In connection with these tests a noteworthy agreement exists between the results noted and those previously obtained from tests of machines of similar design installed in the Manhattan station of the Interborough Rapid Transit Co., New York, and the Long Island City station of the Pennsylvania R. R. At the same loads and with equivalent operating conditions the performance of the machines is almost identical. These economic results, while not exceeding in actual steam consumption to best records of European practice, yet are considered extremely good by the manufacturers, in view of the operating

conditions under which the test was conducted and are held to represent the best results that have yet been obtained by any turbine under the conditions named.

THE FUEL-TESTING PLANT OF THE UNITED STATES GEOLOGICAL SURVEY AT NORFOLK, VA.

By HERBERT M. WILSON, M. Am. Soc. C. E., Chief Engineer of Technologic Branch, U. S. Geological Survey, Washington, D. C.

In April last the Secretary of the Interior organized a new branch of the United States Geological Survey known as the Technologic Branch. Into its charge has been put the work of fuel testing, which was commenced at the St. Louis Exposition in 1904, and the work of structural material testing, which has been in progress at the Forest Park plant at St. Louis since 1905. In addition to the straight fuel testing, investigations of the occurrence of fuels—chiefly on the public lands of the West—are being carried out; investigations into better methods of firing coal, and into the variety of furnace in which certain coals should be utilized, with a view to the abatement of smoke and best utilization of heat; records and investigations into waste of coal due to mining explosions; the causes of the latter and their possible prevention; the better use of explosives and the use of better explosives in coal mining; investigations into the mining engineering features of coal waste, due to bad methods of mining, handling and marketing the product; investigations into the occurrence, preparation and utilization of lignites and of peat as a fuel; and tests of the relative efficiency and commercial value of liquid and mineral fuels as utilized in internal-combustion engines—chiefly gasoline, kerosene and denatured alcohol.

During the past three years the testing of the fuels of the Middle West has been carried on at the St. Louis plant, and a complete survey of the fuels of the United States in the matter of chemical composition and combustion tests has been made in a preliminary way.

In all something less than 200 separate fuel deposits from Pennsylvania to California have been so tested. At the time of the organization of the Technologic Branch it was decided to remove the fuel-testing plant from St. Louis to Norfolk, Va., with a view to making more extended tests on the fuels reaching tidewater for shipment to the naval and merchant marine of the world. These tests are planned especially with a view to indicating the general methods of utilization of the fuels from the New River, George's Creek and Pocahontas coal fields of Maryland, Virginia and West Virginia, in the hope of securing increased efficiency and of preventing waste and of thus extending the life of these coal deposits. This work has especially in view the needs of the Navy of the United States.

The Norfolk plant is located within the grounds of the Jamestown Exposition. This location was selected because of convenience in the way of transportation facilities furnished by the Tidewater Railway, now known as the Virginian Railway, and because the Exposition Company erected a building for the occupancy of the fuel-testing plant, thus saving that expenditure to the Government. This building is situated to the right of the main entrance to the Exposition grounds, near the Service Building, and is admirably suited to the work. It is a one-story building, 325 feet in length by 125 feet in extreme width, and 44 feet in clear maximum height. One end, 75 feet in width, separated from the remainder by a brick fire wall, is devoted to the briquetting plant. A portion of the same wing, enclosed in brick fire walls, is occupied as a boiler room. The main part of the building is occupied by gas and gasoline engines, turbo-generators, chemical laboratories, offices, etc. A portion of the remaining wing is occupied by a section of the structural-materials testing laboratory and by offices; and the remainder of this wing by a commercial exhibit of denatured alcohol appliances in space not immediately required by the Federal testing plant.

Tests of fuels are being made on long runs, on one type of fuel at a time, the tests extending over a sufficient period to

permit of their approaching commercial conditions. For example, gas-producer tests made at the St. Louis laboratory were on runs of about 48 to 60 hours, and the maximum commercial conditions were only barely reached in the operation of the producers when the test would have to be discontinued. The results, while exceedingly valuable as indicating the relative efficiency of fuel in the gas producer or under the steam boiler, are being supplemented by the more prolonged tests being made at the Norfolk plant. Each type of fuel, such as that from a mine typical of a given coal seam is tested in lots of about 150 to 250 tons. A portion is converted into producer gas, and as the test progresses the best method of handling and firing to produce the maximum of high-grade gas is ascertained.

This plant consists of two Taylor pressure-gas producers; the inside lining of the brick jackets is 7 feet high, and the height to the top of the casing 15 feet. The gas from the producer passes through a tar extractor and a scrubber, and then through a purifier into the gas holder. In operating this producer during several years on various types of soft coal, slack, lignite and peat, no difficulty has been found in working effectively, regardless of the amount of volatile matter distilled from the fuel. The gas holder has a capacity of 4,000 cubic feet. From it the gas passes through a meter and operates a Westinghouse gas engine of 235 H.P. The power generated is converted into electrical energy by belting to a 200-kw. Bullock generator, and is measured on a switchboard from which the load can be regulated so as to maintain a steady full-load value.

Among the determinations made in this section are: The length of the test run; the effect of the size of coal; the maximum returns from different fuels; the response of the producer plant to sudden changes in demand for power; the effect of rapid load variations; the best depth of fuel bed, etc. In the course of the test, time is kept of the labor force, of the weight of all fuel, of ashes, of the fuel consumed in the auxiliary boiler, the amount and value of the water consumed, and every item which goes to make up the cost of producing 1 H.P. from a given quantity of fuel.

Every shipment of fuel which is tested for producer gas is also tested for combustion in the steam section; and a portion of it is manufactured into briquettes, which are then tested in comparison with the run-of-mine coal from which they are manufactured, under the same boilers and by identical tests.

The steam section contains two Heine boilers of about 210 H.P. each, and one Babcock & Wilcox marine-type water-tube boiler of 250 H.P. One of the three furnaces is equipped with a Roney overfeed stoker, one with a Jones underfeed stoker, and one for hand stoking only. The three may be operated on natural draft, forced draft, or through the Green Fuel Economizer Co.'s induced-draft fan. The baffling in the different furnaces is arranged differently, so as to procure different lengths of travel of the gases of combustion. It is thus possible to test the same fuel in different sizes, with different methods of stoking and draft, and to baffle in such a manner as to determine the most economical performance under different rates of combustion and the best arrangement of grate and heating surfaces.

The steam generated is used in operating two direct-current DeLaval Turbo-generator sets, each of which is rated at 300 H.P., and is of a speed of about 9,000 r.p.m. The electricity generated, like that generated from the producer gas, is measured and controlled at the switchboard. The work of the steam section is so planned that the performance of the boiler is successfully isolated from the combined performance of the boiler and furnaces, and the tests are being continued with the object of determining still further the efficiency and performance of the furnace alone. The tests are also conducted with a view to determining the method of firing various fuels, so as to reduce to a minimum the smoke emitted from the stack. Both in the producers and in the boiler furnaces holes have been bored at various heights to permit of measuring the highest temperature in various portions of the combustion chambers and fuel beds, and for the extraction of samples of gas for analysis.

Included in the plant are a research chemical laboratory and

a research physical laboratory, for conducting investigations into the gases of combustion. In addition to these research laboratories there is a small commercial gas laboratory for analysis of the gas produced, and a small chemical laboratory for analyzing the fuels and ashes in the steam plant.

The fuels consumed at the plant are inspected at the mine before shipment, and samples are sent to the chemical laboratory for analysis. Similar samples are taken from the car lots as unloaded, and from time to time as the fuel is fed to the producers and the boilers. By this means it is possible to secure, by chemical analysis, data which will permit of determining the values of fuels and their probable performance without test, by comparison with the behavior of those which have been tested and analyzed.

The briquetting room is equipped with a large American briquetting machine of the Renfrow type, and with an English briquetting machine of the Johnson type. The briquettes of the former weigh less than a pound each, are of flattened hemispherical section, and are about the size of furnace anthracite. The briquettes made on the English machine are rectangular, about 3 by 8 inches, and weigh several pounds apiece. Both briquetting machines are operated by a 50-H. P. motor, the fuel being received from overhead bins on a mixing platform; there the coal and the binder (usually about 5 to 6 per cent. water-gas pitch) are mixed, and pass through a disintegrator; they are thence elevated into the tops of the briquetting machines, where they come in contact with live steam at a high degree of superheat. Thence the mixture passes into the briquetting machines, and is pressed into briquettes at pressures ranging from 1,000 to 2,000 pounds per square inch.

Experiments with various binding materials have been made, and are being continued in the experimental laboratory, located at the plant, in which both physical and chemical tests and analyses of briquettes are being conducted. At present the briquetting plant is occupied largely in the manufacture of briquettes from Pocahontas coal, for testing upon naval ves-

sels. One hundred and fifty tons of Pocahontas coal have just been briquetted and delivered at the Navy Yard at Norfolk for testing on a battleship, in comparison with a similar amount of anthracite culm briquetted, and with anthracite run-of-mine and Pocahontas run-of-mine. Similar briquettes are to be furnished for testing under Norment water-tube boilers upon a torpedo vessel, and under Scotch boilers upon a naval tug.

It is believed that these fuels when briquetted will produce a little higher efficiency and develop somewhat more power in a given furnace than when not briquetted, and that they will burn practically without smoke. During the past few months extensive experiments on briquetted fuels, made at this plant, have been conducted in cooperation with the Pennsylvania R. R., the Missouri Pacific Railway, and other roads. The results of the tests made under the direction of Mr. A. W. Gibbs, General Superintendent of Motive Power of the Pennsylvania R. R., have been most gratifying in indications of the probable success to be attained in using briquetted coals, of even comparatively low grade, in producing with a given locomotive a higher efficiency than can be produced by run-of-mine coal; and also in the elimination of smoke by their use, under normal conditions of running.

In the liquid-fuel section, work of much importance is in progress in the determination of the relative availability and efficiency of gasoline, kerosene and alcohol. The equipment includes two 15-H.P. Otto gasoline engines and two 15-H.P. Nash gasoline engines; there are also a 2-H.P. and a 15-H.P. International Harvester Company's gasoline engine. Experiments are being conducted on the relative performances of gasoline and of kerosene, and of both as compared with alcohol. The experiments cover the whole range of this field, including the examination of the different carburetors with a view to determining the most efficient method of vaporization. As with the other tests, exact record is kept of the amount of each fuel used, the labor and water, etc., so as to determine the exact cost of producing a unit amount of power.

The fuel-testing plant is equipped with modern coal-handling

apparatus. Hopper dumping cars are switched on to a track adjoining the plant, and the fuel is dumped into a pit, whence it is lifted 40 feet by a link-belt elevator; from this it can be discharged either through a chute into the gas-producer bins, which are near by and on a level with the top of the producers, or into the bins of the boiler plant, or on to a Robins conveying belt which delivers the coal into one of two storage bins located under the roof of the briquetting plant.

Tests of lignites, both for producer gas and for briquetting, will be conducted in this plant, which possesses a large German lignite-briquetting machine. This lignite press has, however, not yet been installed, and it will doubtless be some time before funds will be available to permit its operation.

Tests of peat are being made in connection with a complete survey of the peat deposits of the Atlantic Coats. Several carloads of Dismal Swamp peat will be macerated in the disintegrator on the plant, dried and converted into producer gas.

In addition to the probable lines of tests in connection with the survey of the fuel deposits of the United States, which are being conducted at the Norfolk plant, the Technologic Branch of the Geological Survey has at Denver a plant equipped for tests of the availability of the western coals for coking purposes—especially those on Government public lands. This plant is equipped with a washery provided with different types of jigs and with means for float and sink tests, etc.; the washed coal is thence conveyed to two standard beehive coke ovens. It is believed that the extended series of tests being conducted at this plant will result in improving the quality of the output of coke produced from western coals.

—"Engineering News."

CECILIE—THE FASTEST RECIPROCATING—ENGINE LINER AFLOAT.

The arrival at the port of New York of the transatlantic liner *Kronprinzessin Cecilie*, of the North German Lloyd Steamship Company, marks the advent of the last and finest of that great quartette of high-speed ocean steamers of this

company, which has helped so greatly to advance the speed and comfort of transatlantic travel. Commencing with the *Kaiser Wilhelm,* which was the first ship to maintain an average of over 23 knots an hour across the Atlantic, the company have placed in service at intervals of a year or two the *Kronprinz Wilhelm,* with a record of 23.47 knots, and the *Kaiser Wilhelm II,* which raised the speed to 23.58 knots, the present record of the Atlantic. The last-named ship, which was brought out in 1904, proved to be so eminently satisfactory that, when the company decided to build the *Cecilie,* they considered that they could not do better than duplicate the *Kaiser Wilhelm II* in every particular. This was done; and that the ship will equal, and probably exceed, the performance of the sister vessel is shown by the fact that on the trial trip, over a measured course of 60 miles, the *Cecilie* averaged a speed of 24.02 knots.

In view of the fact that the *Lusitania* has shown such good results on her trial trip, and is likely to capture the Atlantic record, and that the German Lloyd Company are certain, in the future, to make an effort to win back the record, it is probable that the *Cecilie* is the last high-speed transatlantic steamer of very great power that will be built with reciprocating engines. In fact, it may be taken that in this ship the German shipbuilders have carried the development of the reciprocating marine engine up to the high-water mark of its possibilities. Further advance will be either along the lines of the steam turbine, or of turbo-motors with the electric motors direct connected to the propeller shafts; or it may be that the next advance will be marked by the introduction on a large scale of the marine producer-gas engine. That the limits of the reciprocating engine have been reached is shown by the great magnitude attained by many of the engine parts, and notably by the propeller shafts. These last in the *Cecilie* are 25¼ inches in diameter, and upon each devolves the heavy duty of transmitting at times as much as 24,000 horsepower. This is an enormous load to be imposed, day and night for nearly a week upon a single member, and the question of further increase of power is halted by the serious difficulties that would be encountered

in the forging of shafting of the necessary size, elasticity and durability. For this reason alone naval designers are being driven to the use of three and even four shafts; and to such an arrangement the steam turbine lends itself perhaps more readily than the reciprocating engine. In the new record-breaking ship which the North German Lloyd's company are pretty certain to undertake, it will be a question of great interest as to which of the four leading types of turbine will be adopted—whether the English Parsons, the American Curtis, the German Zoelly, or the French Rateau.

If we are right in our conjecture that the *Cecilie* will mark the highest point to which the development of the reciprocating engine will be carried in high-powered steamers, the *Cecilie* and the sister ship will always be notable landmarks in the future annals of the marine engine. They have set the figures for fuel economy at a point which must ever tax the skill of the steam turbine builders to surpass, if indeed they ever attain it. Before giving any details of the motive power, we may mention that the *Cecilie* is 706 feet long, with a beam of 72 feet, a depth of 44 feet 2 inches, and a displacement of 26,000 tons. She has the usual double bottom, and the hull is divided into twenty watertight compartments. All the bulkhead doors may be closed directly from the bridge; and like all modern liners of the first class, she may be considered unsinkable by any of the ordinary agencies of disaster. Of the interior accommodations of the ship it is sufficient to say that they are superior, even to those of the later ships of this line; while the decorative features are marked by the simplicity and refinement of the latest school of marine decorative work.

The engines of the *Cecilie* are four in number, each carried in its own separate watertight compartment. Each engine has indicated about 12,000 horsepower in actual service. They are placed in pairs in tandem, two on each shaft. In the four engines there are altogether sixteen cylinders—four high-pressure, four first intermediate, four second intermediate and four low-pressure cylinders. The high-pressure cylinders,

which are carried in tandem over the first intermediates are 37⅜ inches in diameter; the first intermediates are 49¼ inches; the second intermediates, 74⅞ inches in diameter, and the low-pressure cylinders reach the truly enormous size of 112¼ inches diameter, these being the largest marine cylinders, we believe, ever built. The common stroke is 6 feet. Steam is supplied from nineteen cylindrical boilers, at a pressure of 230 pounds to the square inch, through four main steam pipes, each of which is 17 inches in diameter. To produce this steam 764 tons of coal are burnt every twenty-four hours in 124 furnaces; and, as we have above stated, the resulting 48,000 horsepower of the main engines has proved sufficient to drive the vessel at slightly over 24 knots an hour.

It is an interesting coincidence that the finest, and what will probably be the last, of the reciprocating-engine liners, should be placed in service at about the same time as the first of the new high-speed turbine liners. The performance of these two ships will naturally be watched with the keenest interest; for in spite of the fact that the turbine-driven ship greatly exceeds the other in size and power, the *Cecilie,* herself, with her displacement of nearly 30,000 tons, and her horsepower of nearly 50,000, is sufficiently large and powerful to eliminate, when driving into a head sea, much of the advantage due to the higher momentum of the larger ship. It is to be hoped that the facts as to relative coal consumption will be made available for the technical world. Just now, the steam turbine is being saddled with a reputation for extreme costliness in fuel; and predictions along these lines are being made so freely regarding the new Cunarders, as to suggest that perhaps the wish, in some quarters, may be father to the thought. For ourselves, we think it is likely that, because of the size of the plant and the high speed at which the turbines are being run, the new Cunarders will show about the same economy as the latest reciprocating-engine ships. This has been brought down in the *Cecilie* to 1.4 pounds of coal per horsepower per hour, including the auxiliaries.—"Scientific American."

THE TURBINE, AND GERMAN LINES.

The German lines are by no means convinced that the turbine is the best form of steamer for oversea purposes. Director Wiegand of the North German Lloyd says that the *Lusitania's* victory is not to be attributed to the fact that she has turbines but that she has such high horsepower—70,000.

The director continues: "The *Lusitania's* turbines, with their relatively small number of revolutions, should not run more economically, to say the least, than a good triple or quadruple-expansion steam engine. The economic value of the turbine lies chiefly in its high revolutions and the satisfactory utilization of steam thus secured. The rapid revolution of the propellers, however, means a reduction of their efficiency. In order to make the propellers revolve at the speed which renders them most effective, it is necessary to reduce the speed of the turbine to a point where it no longer operates economically. After many years' experience it has been found that the propellers of the North-German Lloyd's express steamers work most economically at about 80 revolutions per minute, whereas the *Lusitania's* screws have 180 revolutions. For this reason I do not believe that the *Lusitania's* machinery plant represents an advantage over the prevalent type. In other words, if I build a ship of the same dimensions and put into it an equal power in the form of reciprocating engines, the effective result will in all probability be the same, if not superior to the turbine-equipped steamer. One disadvantage of the turbine steamer is that it possesses inferior maneuvering capacity, because all the steam at hand cannot be used for backing.

"Another question arises: What practical advantage is it to increase the speed of steamers in the North Atlantic traffic to 24 or 25 knots per hour? I am of the opinion that the advantage is just about equal to zero. With our present speed of 22 to 23 knots we arrive at New York by day, whereas we should arrive at night if our steamers should make 24 or 25 knots. Precisely for this reason it was that we brought our express steamers up to 22 and 23 knots. It is well known that we refused to accept the *Kaiser Friedrich* several years ago be-

cause it did not make contract time and would land in New York at night. To bring us any practical advantage in this respect, therefore, the speed of a steamer must not stop with 24 or 25 knots, but would have to make 25½ to 26 knots. That is a circumstance which is not sufficiently taken account of in Germany. We are under obligations to call at Southampton and Cherbourg within certain hours in order to make connections with the trains for London and Paris, and must accordingly stop our steamers at the usual hours, and we could not equalize matters (in case very fast steamers were used) by postponing the sailings for several hours."

Director Wiegand concludes that if the Lloyd should increase the speed of its steamers to 24½ knots their operation would be less economical, since the vessels themselves would cost more, consume more coal, and besides all this would arrive in New York at night, after which they would be compelled to lie at their docks till the regular time for sailing, thus counterbalancing any advantage from greater speed. "Of course," he adds, "if we increase their speed to above 25 knots they would arrive at comfortable hours, but the cost of such a steamer of about $7,500,000, along with the vastly increased expense of operation, would be out of all proportion, economically, to the saving of time at sea. Such a saving would not present a strong enough attraction for the general public to make them willing to pay the increased cost of tickets. Hence, as our company is dependent upon its own resources (instead of subsidies) it could not get any business advantage from putting on such fast steamers."

The report that the vibration on the *Lusitania* is considerably greater than on the fast German steamers has been looked upon here as an additional reason for the German lines to adhere to the old type of engine. Owing in part to that increased vibration the Germans think that this steamer and the *Mauretania* will not prove dangerous competitors for the first-class passenger traffic between Europe and New York. Only when they make much faster time than the *Lusitania* did on her maiden voyage could they become dangerous rivals; but then

the vibration would increase still more annoyingly, so as to neutralize the attraction of quick time; for comfort is, after all, the chief thing desired by the ocean traveler, and a day more or less at sea cuts a very slight figure with everybody except a few hurried business men.

VENTILATION AND REFRIGERATION OF AMMUNITION HOLDS.

Translation of abstract of paper read at the Bordeaux International Congress in Naval Architecture by ADRIEN BOCHET.

The safety of ammunition holds has always engaged the attention of naval architects, and for many years past measures have been taken for isolating these holds, for protecting them against the various causes which may lead to an accident, and for inundating them in the event of danger. Modern powders, and the considerable development in the use of machinery on board ship, have increased the risks in a large proportion. Modern powders have excellent ballistic properties, but are also most unstable, and their instability increases very rapidly with an increase in temperature. By their gradual alteration they also set free a quantity of inflammable gases which may give rise to explosive mixtures. The development in engines and boilers, in auxiliary engines, and in the extent of steam pipes laid throughout the ship, has led to a greater heating of the various compartments. The distribution of the armament, and the necessity of providing ammunition holds in proximity to the guns to be served, have often resulted in the holds being in locations that are particularly unsuitable from the point of view of temperature. Therefore, concurrently with a gradual increase in sensibility to heat, which forms one of the characteristics of modern powders, causes tending to augment the temperature of the ammunition have increased in a large proportion also, and numerous attempts have been made with a view to ensure the artificial refrigeration of the holds.

Cooling by ventilation alone is a simple and safe means to prevent the accumulation of explosive gases, but is quite

insufficient to ensure a decrease in the temperature of the holds when the temperature of the air outside reaches 20 degrees (say 70 degrees F.), and when the causes making for an increase of temperature in the holds attain a certain importance. The heating of the air which enters the ship occurs rapidly on contact with the warm bulkheads, by reason of its low specific heat. A rise of 10 degrees C. has often been noted for courses which appeared as short and direct as possible between the upper works of the ship and the fans placed in the holds. Cooling by the means resorted to in refrigeration holds, such as those for the transport of meat, appeared at first to constitute a final solution of the problem. But this failed completely, owing to the very great difference existing between these holds and those for storing ammunition. In the former there is no ventilation, and they remain closed during the whole passage. They are also maintained at a very low temperature, and one result of this is that the small proportion of moisture contained in the local air on starting gets deposited on the products contained in the refrigerating chamber, and does not injure them in any way. Ammunition holds, on the other hand, require ventilating. Even were one so imprudent as to do away with ventilation completely, or to reduce it in too large a measure, the requirements of the service would demand the frequent opening of the hold, thus allowing every time the outside air to penetrate into it. The temperature which it is suitable to maintain in an ammunition hold is much above 0 degree C. (32 degrees F.). One result of this frequent renewal of the air in an ammunition hold is that the moisture it contains is condensed on all the surfaces which are maintained at a lower degree of temperature than that of the outside air, and the water which cannot be congealed, as in the case of a provision hold, trickles down the partitions of the holds and the ammunition. This very grave disadvantage can only be avoided by cooling the air entering the ammunition hold, so that it is not at a higher temperature than that of the contents or of the walls of the hold.

The only rational means to cool ammunition holds is to ven-

tilate them with air suitably cooled, and this deduction has received the sanction of actual practice, as it is the only method of cooling which has prevailed so far. It has been applied to fourteen French battleships, which carry together forty-three ammunition-hold cooling plants, and to eight Russian battleships, which are fitted with thirty-seven similar plants. It will be fitted also to the armored cruisers *Waldeck-Rousseau* and *Michelet,* now in course of construction. The ammunition holds of these two ships are to be provided with aero refrigerators, in which a circulation of artificially-cooled liquid is maintained constant; the dynamo rooms are to be cooled by a similar installation, but with sea-water circulation.

Until recently, the maximum temperature which was thought advisable for ammunition holds was 35 degrees C. (95 degrees F.); now, however, this limit has been brought down to 30 degrees C. (86 degrees F.). With the former limit direct cooling by sea-water circulation sufficed, as the temperature of the water taken from a certain depth remained perceptibly lower than 35 degrees C. in all parts of the world. The latter limit, however, can only be reached by having recourse to artificial refrigeration.

The refrigerators for cooling the air consist of metallic surfaces, on one side of which the cooling liquid circulates, the air to be cooled circulating on the opposite side. A pump ensures the circulation of the liquid, and a fan that of the air. The complete apparatus are built by F. Fouche, 38, Rue des Ecluses St. Martin, Paris.

The extent to which refrigeration has to be carried depends upon the amount of heat which enters the hold: it is necessary, in the first place, to reduce to the lowest possible minimum this amount of heat. The afflux of heat in the hold is caused by radiation from the warm sides and by conductivity from the metallic pieces they contain, and which are connected to the heated sections of the ship. The methods of obtaining a thermic insulation of the hold are the following: Radiation from the warm sides can be reduced by an inside lining made of substances the conductive and emissive properties of which

are low, such as cork and asbestos. Another solution is to build a double wall, to obtain an air lining, in which case it is advantageous to ensure in the double wall a circulation of air at the lowest possible temperature. The circulation of a cold liquid inside the double wall will give most satisfactory results, provided the liquid be brought sufficiently in contact with all the metallic pieces that are liable to carry heat into the hold by conductivity.

It is easy in principle to ensure the thermic insulation of the hold, but in actual practice the application of the various methods is surrounded with great difficulties, owing to the arrangement of the holds, the small space available, and the necessity of preventing every cause of damage to the ammunition. The use of simple insulating inside coverings of a thickness proportional to the heating of the walls and the available space is evidently the most easy solution of the problem. As this method has hitherto proved of sufficient efficiency, it has alone been developed in actual practice. The use of double walls with air circulation has, however, been successfully combined with the insulation of the sides.

The use of a cold-liquid screen is surrounded by the following difficulties: In order to be efficient, there should be no break or interval in the current. It should reach the whole of the metallic pieces which are liable to cause heat in the hold by conductivity. All risks of inundating the hold in case of leakages have to be removed. The cold screen is constituted by a double wall, but it is absolutely impossible to give the space sufficient dimensions to allow a man to enter it in order to paint the plates and keep them in a good state of repair; this proscribing the use of all liquids, such as sea water, which may corrode the plates. The solution consists in causing to circulate inside the double wall a liquid, such as fresh water charged with oil, or milk of lime, with which there is no risk of corrosion. The total quantity of liquid thus circulating has to be as low as possible, so as to prevent in any case the risk of inundation of the hold. If a leak did occur, a small quantity of liquid only would escape without risk of damaging the ammunition. Sim-

ple means suffice to show when an accident of this kind has happened.

The paper was accompanied by drawings showing how a hold could be insulated and the method followed for ventilating it with cool air. The insulation of the hold is completed by the flowing of the exhausted air round the walls. The liquid and the air for ventilation can be cooled simply by sea water or by means of a refrigerating machine, according to the temperature required to be maintained in the hold.

The amount of heat Q^1 penetrating through the walls is proportional to their surface $\int s$, to the difference of temperature between the two sides $T—t$ and to the time.

The number of heat units going through a given partition per unit of surface and time q^1, for one degree difference in temperature, is determined experimentally. Numerous tests have established the figures corresponding to partitions covered with the usual insulating material. The following formula is obtained:

$$Q^1 = \int s\,(T = t), q^1$$

for the unit of time.

Further, the metallic parts, such as partitions, bulbs, standards, projectile supports riveted to a warm wall, cause a quantity of heat to penetrate the hold; this is

$$Q^2 = \int s \frac{q^2}{e}\,(T - t),$$

where s is the section of the metallic part considered; e the distance between the point where the heat maintained at temperature T penetrates, and the point whence this heat is emitted inside the hold, the temperature of which is t; q^2 being the number of heat units passing per unit of time and per unit of section of the piece, considered between two points, with the unit of length between them, and maintained at a temperature difference of one degree.

Neglecting the amount of heat brought in by the personnel,

caused by the handling of the projectiles, also the entrance of warm air, the total quantity of heat penetrating the hold will be

$$Q = Q^1 + Q^2$$

This amount of heat has to be totally carried away by ventilation in order that the hold may be maintained at the required temperature. It is necessary, therefore, to send into the hold a volume of air V capable of absorbing the quantity of heat Q, by being heated from its entrance temperature τ to an outlet temperature θ. The calorific capacity of air being 0.3 heat unit per cubic meter, the following will be the formula:

$$V = \frac{Q}{0.3\,(\theta - \tau)}.$$

It is necessary to ascertain that the volume of air thus determined meets the ventilating conditions of the hold according to the nature of ammunition it contains.

In order to bring the volume of air V from the initial temperature θ measured at the moment it enters the refrigerating machine to the temperature θ measured at the outlet, the amount of heat given up by the air proper, by the water vapor it will contain after cooling, and that given up by the vapor which condenses, have to be deducted.

The specific heat of dry air being 0.3 heat unit (calorie) per cubic meter,

That of vapor being 0.48 heat unit (calorie) per kilogram.

The latent heat of water evaporation being 606.5 heat units (calories) per kilogram,

p the weight of water vapor contained in one cubic meter of air drawn, and

p^1 that remaining in one cubic meter of the air after cooling

The formula for the heat to be absorbed will be

$$Q^1 = V[\theta - \theta\,(0.3 + 0.48p^1 + 606.7\,(p - p^1)].$$

In designing the refrigerating machine it is necessary to add to this amount of heat Q^1 that resulting from the heating of the apparatus themselves, and of the pipes in which cold air and liquid circulate.

The dimensions of the apparatus and of its auxiliary parts—the circulation pump and the fan—are deduced from the volume of air V flowing per unit of time, and from that of the cold liquid necessary to carry away the quantity of heat Q^1 increased as aforesaid.

It should be remarked that the power of transmission of heat of an aero-refrigerator from air to water is, so to speak, without limits; it depends solely upon the output. It increases with the speed, so that the dimensions of the apparatus, suitably proportioned, are solely dependent upon the required pressure of the air and the cooling liquid.

Observations made on a number of installations show that the machines supply air easily at a temperature which does not exceed by much more than about 1 degree C. that of the cooling liquid, whatever be the temperature of the air drawn into the machine. The water-vapor contained in the air is completely condensed in the aero-refrigerator in a proportion which corresponds with the fall in temperature. The hold thus remains perfectly dry.

Tests made on board the *Sully*, in the Far East, have shown that with temperatures of 31 degrees C. (87.8 degrees F.) for sea water, and 36 degrees C. (96.8 degrees F.) for the air, the apparatus maintained the hold at a temperature of 32 to 34 degrees C. (89.6 to 93.2 degrees F.).

The results of other tests, and illustrations of the apparatus, were given in the paper.—"The Marine Review."

NEW FOURTEEN-INCH GUNS FOR COAST DEFENSE.

By CHARLES A. SIDMAN.

Plans have been perfected for the new 14-inch caliber guns which are to be added to the existing coast-defense system now in place along the coast from Maine to Washington State, and Gen. Crozier, Chief of Ordnance, United States Army, has the Watervliet arsenal engaged in the manufacture of these new pieces.

The 14-inch gun will be something new in ordnance design, and although fully two inches larger in caliber than the standard coast-defense gun of the first grade (a 12-inch caliber), the new gun will be shorter in length, and the outside diameter will be smaller. The powder chamber will be less than the gun now in use, and it will be lighter in weight.

It is proposed to make a weapon that will have a range and striking force at least equal to the present standard 12-inch gun, but which shall greatly exceed the very limited life of that gun. One strong point also in favor of the new gun is the fact that its first cost will be less than that of the 12-inch gun, while the addition to the life of the piece will result in a greater economy.

In speaking of the new gun, Gen. Crozier said:

"There will be a vast difference between the present standard 12-inch gun and the new 14-inch gun. The 12-inch gun fires a projectile of 1,000 pounds weight, with a velocity of 2,550 feet per second, using 366 pounds of powder, and only has a life of between sixty and seventy rounds, before it has to be relined. This necessitates dismantling and shipping back to the factory for that purpose. The new gun, which is much shorter in length than the 12-inch gun, will fire a 1,600-pound projectile, use nearly 100 pounds of powder less than the 12-inch gun, and only gives a muzzle velocity of 2,150 feet per second, while its life will be nearly four times that of the present standard.

"By reason of the lower velocity required and the consequent smaller charge, it is possible to make the 14-inch gun proportionately shorter than the 12-inch gun, and the smaller charge of powder also involves a less diameter of powder chamber, and, therefore, with the same thickness of wall of the chamber in caliber, a less exterior diameter of the gun over the breech. These elements of saving are so considerable that the weight of the 14-inch becomes actually less than that of the 12-inch; and as a lower maximum pressure is needed, it is possible to attain all the strength which will be used without employing the most expensive steel.

"The muzzle energy of the 14-inch projectile will be about 15 per cent. greater than that of the 12-inch; and because of its lower velocity and its greater weight, the retarding influence of the air will be much less upon this projectile, so that the gain of energy will be in greater proportion with each increment of range. Because of its lighter weight and of the cheaper material of construction, the cost of the new gun will be less than that of the 12-inch gun, while the cost of powder will also be less than that for the 12-inch by about $70.

"The cost of the 14-inch armor-piercing shot will be about $100 more than that of the 12-inch shot, so that the total cost of a single round will be about $30 greater. Taking into consideration, however, the rapid deterioration of the 12-inch gun, and adding the cost of relining to that of the ammunition, which would correspond to the number of rounds making relining necessary, the cost per round, including the deterioration, for the 14-inch gun is only about 68 per cent. of that for the 12-inch gun.

"The penetration of the 14-inch projectile through Krupp armor at 10,000 yards is about 11 inches, while that of the 12-inch is about 10½ inches; while the range at which the 14-inch projectile will penetrate 12-inch Krupp armor is about 8,700 yards, as against 8,500 yards for the 12-inch projectile.

"To sum up, then, it appears that in situations requiring the greatest power, a 14-inch gun, with 2,150 feet per second muzzle velocity of projectile, instead of the 12-inch gun with 2,550 feet per second initial velocity, gives us a lighter gun, a cheaper gun, a heavier projectile, greater muzzle energy, a still greater proportion of energy at each distance beyond the muzzle, and a life four times as long."

Regarding the life of the 12-inch and 14-inch guns, it has been considered that, in attempting to run by fortifications guarding the entrance of a harbor, the period that would elapse from the time that the leading vessel of the fleet would come within range until the last vessel would pass beyond the range of the coast guns would be about two hours. It is therefore evident that a new 12-inch gun would not last through such an

engagement; and considering that this gun is capable of firing for a considerable interval at the rate of forty-five rounds per hour, it is seen that the limit of its life would be reached in less than two hours. With the 14-inch gun, the life of the gun before its accuracy would show impairment would be about 240 rounds, corresponding to about six and a-half hours of continuous firing at the rate at which it is thought it can be fired. The weight of the new gun will depend greatly upon the manner in which it is built. It is yet to be decided whether the guns shall be of the ordinary built-up forged-steel type or shall be of the more modern wire-wound construction. If wire-wound, the weight will be about 110,500 pounds, while the built-up type will weight 12,500 pounds more. The present standard gun weighs about 1,500 pounds more than the average for the new gun. They will be mounted on the disappearing carriage, and will be installed wherever needed.—"Scientific American."

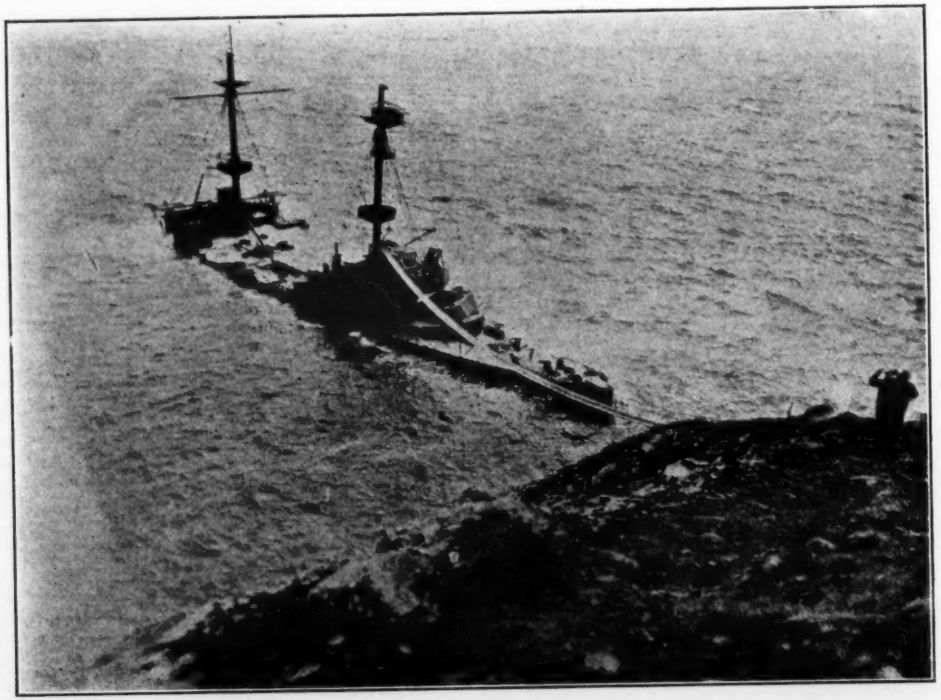
BREAKING UP THE ILL-FATED BRITISH BATTLESHIP *MONTAGU.*

By HAROLD J. SHEPSTONE.

The first-class British battleship *Montagu,* which went ashore on the southwest corner of Lundy Island, in the British Channel, in a dense fog on May 30, 1906, is now being broken up piecemeal by the shipbreakers, and gradually the huge warship is disappearing.

The gallant attempt made to salve the vessel, although a complete failure, was nevertheless a daring piece of work deserving of the highest praise. For months the Liverpool Salvalge Association, and the most skilled workmen from the British dockyards, with the most up-to-date tackle at their disposal, endeavored to lighten the ship and get her off the rocks, but without success. The whole history of the operations was one steady and persistent fight against unsurmountable difficulties.

Twenty-four hours after the vessel grounded, practically every compartment in the ship was full of water. The cap-



THE WRECK OF THE "MONTAGU" AS IT APPEARS TODAY OFF LUNDY ISLAND.

stan-engine room forward, the compartments under the forward 12-inch turret, all the boiler rooms, the starboard engine room, and the steering compartments aft, were open to the sea, the water rising and falling with the tide. The port engine room was tight, but had to be flooded by opening the sluice valve between the two engine rooms to prevent the ship taking a heavy list. At high tide the water rose about two feet above the upper deck at its lowest part, while at low tide the water fell to about seven feet below the usual waterline of the ship.

The vessel rested on a rocky bottom, the general surface being fairly level, but with many pinnacle rocks. It was discovered by divers that the damage to the bottom was very serious. A large rock penetrated about ten feet into the ship under the capstan engine forward. Along the starboard side the bilge keel was more or less carried away, and several large holes were discovered. The port propeller, A bracket, and shaft had disappeared completely; the starboard A bracket was cracked, and one blade of the starboard propeller was carried away, and the lower parts of the sternpost and rudder were broken off. To repair the vessel and pump out the water so that she could be refloated was the task the British Admiralty assigned to Admiral Wilson, and every possible assistance was rendered him in his work.

After a complete examination of the ship it was decided to pump the water out of the vessel as far down as possible by pumps driven by compressed air. Before this was done the battleship was lightened by removing the 6-inch and smaller guns, torpedo nets, chain cables, etc. Owing to the surrounding rocks and shallow water, only lighters or very small vessels were able to approach to the wreck. This added greatly to the difficulties of the operations, as in most cases heavy weights, such as boilers, etc., had to be first hoisted into lighters. One of these craft, containing four 6-inch guns and several torpedo nets, in being towed was carried by the current over the rocks and foundered. She was, however, successfully salved some weeks later.

Within ten days from the disaster the following centrifugal

pumps were in action: One 12-inch, three 10-inch, three 8-inch, one 5-inch, and one 3-inch, and an additional 12-inch pump was available on board one of the salvage boats supplied with steam from her own boiler. Three more 10-inch pumps and two 8-inch pumps were also in course of erection. A week later seven more pumps had been fitted up on board. They had a total pumping capacity of 8,600 tons per hour. Their object was to command the different compartments in which they were placed, and generally clear the water down to about two feet above the platform deck. Owing to the size of the rose boxes they could not clear the water closer to the deck, and efforts were then directed to find compartments below this level which were tight, and which could be used as wells for pumping suctions. In the fore end of the ship this scheme had to be given us as hopeless, though it was partially successful in the after part of the vessel. Here a U-shaped pocket or well was lowered through the hatch of the submerged torpedo room, and guided into place under the deck and shored up by divers.

Although it was found that in many ways the minute subdivision of the ship helped the salvage work, it made the work of pumping the ship clear of water more difficult. To prevent the vessel bumping at high water it was generally necessary to let her fill up with water then, and arrangements had to be made to pump out the various compartments rapidly, say in about eight hours. It was not practicable to provide special pumps or suctions for all the minor compartments, and time did not permit of the suction pipes being moved from one compartment to another. Drain holes had therefore to be cut —mostly under water—to enable the smaller compartments to drain into the larger ones, where pump suctions were fitted.

The next task was the covering over of the hatches to the engine and boiler rooms at the elevation of the main deck by plates. Then a number of camels, or tanks, were fixed along the side of certain portions of the ship. By experiments it was estimated that to float the vessel safely, a plant capable of producing 6,000 cubic feet of air per minute would be required

to drive the water out of the ship. It was obtained, but failed to respond to the gallant efforts of the salvage operators.

On August 6 last the British Admiralty, after nearly fifteen months' strenuous efforts to refloat the warship, decided that the task was hopeless and abandoned it. Before doing so the vessel's heavy guns, a large quantity of her armor plates, boilers, machinery, and chain fittings, were removed. The wreck was then purchased by a South Wales syndicate for a lump sum, who hope to secure the valuable steel and iron composing the hull.

The syndicate at once established quarters on the island, where they have a staff of some fifty experienced workmen and a couple of small steamers and lighters. Their engineers have thrown an aerial footway, over 500 yards in length, from the top of Lundy's precipitous cliffs to the roof of the chart house on the wreck. Down this footway the shipbreakers pass to and from their work. The main deck of the ship is always awash at high tide; therefore work can only be carried on for a short time daily. When weather permits, lighters are brought alongside the battleship, immediately below the footway, and piece by piece huge sections of the armor plating are being removed from the huge carcass and transferred to the shore. The amount of work which has been done on the hull will be gathered from the huge "bite" which has been taken out of the bows on the port or seaward side, but it will be many months yet before the *Montagu* has been completely broken up.

One result of this wreck will probably be that the British Admiralty will establish a salvage corps of its own. Hitherto it has had to depend upon outside assistance, whenever any of its vessels have met with serious disaster. So far the British Admiralty have not stated what the salvage operations have cost, but it is estimated to amount to no less than $500,000, while as to the ship herself, she was built so recently as 1903, was of 14,000 tons, and cost the British nation close upon $5,000,000. What the syndicate paid for the abandoned wreck has not been disclosed, but probably only a few thousand dollars."—"Scientific American."

MATERIALS FOR THE CONTROL OF SUPERHEATED STEAM.

A paper presented at the Indianapolis meeting (May, 1907) of the American Society of Mechanical Engineers, by M. W. KELLOGG, '94, M. E.

GENERAL.

Since the introduction of superheated steam as an important factor in economy in stationary power-plant operation, the question of what type of material is best for the proper controlling of the resulting high temperatures has caused a great deal of investigation and interest.

In the following discussion of materials some reasons will be given which are the results of experience and test, and other facts which we have accumulated from reliable sources will be shown. This article treats particularly of what might be called in a general way piping systems, which systems are made up of pipe, fittings, valves and the necessary details connected therewith, such as joints, gaskets, etc., and are here taken up separately.

PIPE.

There can be little question as to the matter of pipe except quality. Of course, welded wrought-iron or steel pipe is successful, but the difference in the quality of pipe under different conditions is very material. As in nearly all instances in a superheated-steam station the old-fashioned screwed joint is not satisfactory, it is necessary to do what is termed "work" the pipe—that is, weld, van stone, etc.—to make either a welded, van-stone, or other joint of the same general description.

The accompanying cut is what is known as a "van-stone joint." For this work, the pipe made from open-hearth steel is a great deal the best for manufacturing reasons, because it can be properly "worked," there being less carbon and the quality being much more uniform. Bessemer-steel pipe will very often act in a satisfactory manner, but one is never sure that Bessemer will run even, and, therefore, troubles may result.

It is practically impossible to "work" wrought-iron pipe. In making what is known as a "van-stone joint," the pipe is

nearly sure to split very badly, not only at the weld but all around its outer circumference.

Nearly opposite qualities from those used for getting good results from "working" pipe are required for threading. A good quality of wrought iron will cut and thread more easily with standard pipe machines and standard dies than a steel pipe, and a Bessemer-steel pipe will thread much more easily with standard dies than open-hearth steel will.

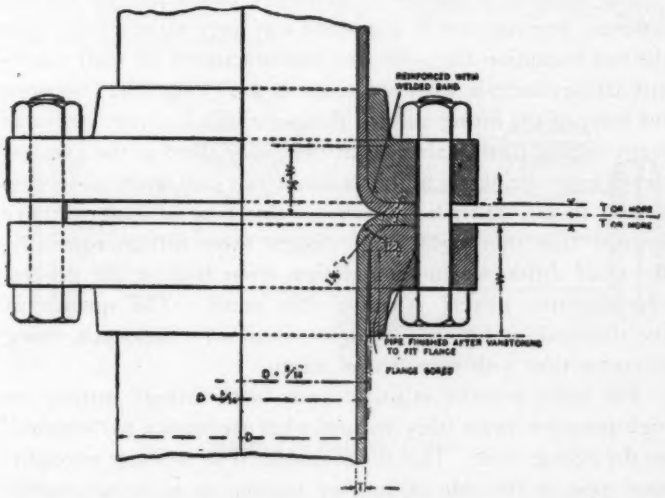


Fig. 1.—VAN STONE JOINT.

A great many manufacturers have difficulties in threading open-hearth steel pipe, for the reason that they set the dies exactly the same as if they were cutting other qualities. This causes ripping of threads, etc. The die in a pipe machine should be set at a greater angle, with the radius of the pipe passing through the point of contact of the die for soft steel than it would be for other kinds, and this in itself will very often eliminate great troubles in this line. The question of lubrication, etc., is also important in this particular.

The ordinary commercial pipe will stand more pressure than

the average person believes. A standard one-inch piece of welded pipe will usually not break under 1,600 pounds per square inch hydraulic pressure. Full-weight pipe I believe to be perfectly suitable for any temperature and any working pressure up to 225 or 250 pounds, as long as it is not thinned at any point by cutting and threading.

FITTINGS.

The designs of fittings as generally manufactured for the different purposes are in a general way very satisfactory, with the one exception that very few manufacturers on their standard articles include what is known as the "long fillet" between the body of the fitting and the flange. This is a very desirable point, owing to the fact that at this place there is the greatest strain from shrinkage in the molds, which also tends to develop porous spots. Most large users of this type of material have learned this thoroughly and design their fittings specially, the chief difference in their design from that of the general manufacturer merely covering this point. The quality of the material in fittings, however, is a very important thing in connection with superheated steam.

The latest practice is to do away with fittings entirely on high-pressure steam lines and put what are known as "nozzles" on the piping itself. This is accomplished by welding wrought-steel pipe on the side of another section, so as to accomplish the same result as a fitting. In this way rolled or cast-steel flanges and a van-stone or welded joint can be used. This method has three distinct advantages, to wit:

(*a*) The quality of the metal used, for reasons explained hereafter when the subject of the effect of heat on metals is taken up.

(*b*) The lightening of the entire work.

(*c*) The doing away with a great many joints.

As a general average, at least 50 per cent. of the joints can be left out, and sometimes this proportion runs up as high as 60 or 70 per cent., according to the layout of the system. If this method is employed substantial welds must be made, not

only to stand the pressures required but also the strains; this is accomplished successfully in Germany, England and the United States.

VALVES.

It is important to have a good design of valve. I believe that nearly all of the designs made by the good manufacturers are entirely suitable, such as a broken or solid wedge valve of the ordinary type, under the condition that all machine work is done thoroughly and the quality of metal used is satisfactory for the purpose intended.

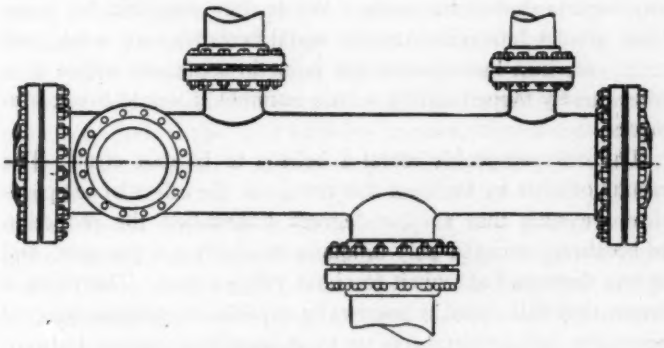


Fig. 2.—WELDED WORK.

It may be interesting to note here the effect of a large range of temperatures on a short piece of steel. By calculation, a piece of steel six inches long, heated 500 degrees, will expand 0.019 inch. This figure is put down to show how variations in the coefficient of expansion of metals by heat have a large effect on the permanency of a valve staying tight, and it can readily be seen that a small part of the distance given is sufficient to cause trouble.

METALS.

I find that different authorities vary slightly in their statement as to what temperatures different metals will stand with good results. German authorities state that cast iron

should not be used above 480 degrees Fahrenheit. Other authorities allow us to go as high as 575 degrees Fahrenheit. Above these temperatures in cast iron the limit of elasticity is reached with a pressure varying from 140 to 175 pounds. Under such conditions the material is strained and does not resume its former shape, and eventually shows surface cracks, which continue to grow until it lets go. These temperatures and pressures also lead in time to a shrinkage of all parts, and to a structural alteration of the metal, which results in leakages in valves at the seatings. Therefore it would seem that iron castings are unsuitable for both fittings and valves to be used in any superheated-steam work. While they may last for some time, after a few years' use the metal becomes very weak, and some cast iron has reached the point in weakness where if it were merely tapped lightly with a hammer it would break into pieces.

The only adaptable metal I believe to be cast steel. The results of tests by Bach on this metal for the effect of temperature are such that at 572 degrees Fahrenheit the reduction in breaking strength only amounts to about 1.1 per cent. and at 752 degrees Fahrenheit to about 7.8 per cent. Therefore it seems that this metal is practically capable of withstanding all pressures and temperatures up to at least 800 degrees Fahrenheit, without showing any appreciable weakness.

The influence of high temperatures on bronze, etc., is very material. At ordinary temperatures this metal has a breaking strength of about 34,100 pounds per square inch and an elongation of 36 per cent. At 572 degrees Fahrenheit the breaking strength falls to about 19,500 pounds per square inch and the elongation to 11.5 per cent. At 662 degrees Fahrenheit, which is quite a common temperature, as the steam leaves the superheaters, the breaking strength of bronze only amounts to 12,200 pounds per square inch and the elongation at the breaking point is only approximately 1¾ per cent. This seems to eliminate entirely brass or bronze of ordinary composition for use with highly superheated steam.

The effect of temperature on nickel is very similar to that

on cast steel, and in consequence this material is very suitable for use in connection with highly superheated steam. Bach recommends that bronze alloys be done away with for use on steam lines above a temperature of about 390 degrees Fahrenheit. Even neglecting the special quality of nickel seatings, on account of the great toughness of this metal and the methods which can be used for securing rings of this substance to the valves and conical surfaces, it has the special advantage of having the coefficient of contraction and expansion with temperature almost exactly the same as that of cast steel, so that no slackness of the rings occurs and the valves remain absolutely steam tight. There are instances in which valves constructed with nickel seatings have been satisfactorily used with steam temperatures up as high as 932 degrees Fahrenheit.

Seats, discs and bushings made of brass or plain bronze do not retain their shape. For spindles on superheated-steam work I strongly recommend nickel-steel, which holds its shape and does not deteriorate with high temperatures. Seatings in valves should not only be screwed in but also pinned in addition, using a fine thread which is very long, to give a tight joint. Seats should also have a flange on the top that makes a joint with the body when screwed down, which prevents the tendency to leak through.

JOINTS.

I think it is generally acknowledged that the old-fashioned screw joint, no matter how well made, would not be suitable for superheated steam work. This leaves for discussion two general types, viz: welded joints, and what are generally known as van-stone or climax joints; that is, any joint where the pipe is turned over the face of the flanges.

In welding a flange on a piece of pipe great care must be taken to see that the weld is perfect, because of the unequal thicknesses of the metal to be so welded. If the weld is thoroughly made this type of joint is very good, although for erection purposes, owing to the fact that the flanges cannot swivel, it does not equal the turned-over joint, as mentioned

above. The manufacturing expenses in making a welded joint are also much more for the same type of work accomplished, on account of the necessity of doing all finishing work after all rough work, such as welding and bending, has been completed. Therefore the cost of welded joints is greater, not only for the work done but because of the increased expense in finishing on account of the necessity of employing methods different from those where the flanges, etc., were all finished before the joints were made, as is possible on the turned-over joint mentioned above.

In regard to the turned-over or van-stone joint, the quality of its manufacture seems to us the most important feature. This joint can be made in a careless way where the pipe is in no way thickened up and only faced on the front. A joint of this kind does not give good results, principally for two reasons:

(*a*) The thinness of the metal on the turned-over portion; and

(*b*) On account of the recesses left between the back of the pipe on the turned-over portion and the flange, owing to the pipe not being finished at this point.

The writer believes, however, that if this point is properly made it is equal to the welded joint as a manufactured article and superior to the welded joint as an article for erection.

To have this type well made the pipe on the end should be thickened up in an amount so great that after the joint is turned over there will be enough metal left to face the turned-over portion on the front, on the outer edge and on the back. We of course take for granted that the flanges are finished on the front. After the work above mentioned is done, the pipe should be as thick on the turned-over portion as the original thickness, or very close to it. Increasing the thickness of the pipe on the end before going through the operation is done in several ways. In a general way, I consider any of the methods satisfactory. The point made of facing the turned-over portion of the pipe on the back is an exceedingly important one, much more so than most people seem to

realize. I have known instances where it has been found impossible to make a ground joint, for no other reason.

In reference to making up a joint, I believe that the face of all flanges or pipe where a joint should be made ought to be given a fine tool finish and have the face level, and that then a gasket of some description should be used. A perfectly made ground joint is a good thing but it is very expensive, and it is hard to get the average contractor to furnish it in a perfectly workmanlike manner. Also, after it is so done, it is liable not to stay tight, on account of the tremendous expansion and contraction causing such strains that the joints are liable to open up, particularly when the pressure is taken off the plant. The simple expansion and contraction on the bolts that make up a joint would cause this.

GASKETS.

There are large numbers of gaskets manufactured of all types and descriptions. It is very hard to take up this subject and be fair to each of the manufacturers, for the reason that practically no one can and has had experience with every type made, so as to judge for himself, and hearsay would lead us to suppose that all of them are at one time perfect and at other times useless.

I have used a great many different types of gaskets, however, and have obtained the best results with a corrugated soft Swedish steel gasket with "Smooth-on" applied, and with the McKim gasket, which is of copper or bronze surrounding asbestos. The ordinary corrugated copper gasket is a very popular make and has been used a great deal. On superheated steam, usually sad results follow. There seems to be some peculiar action that causes this, as on superheated-steam lines a corrugated copper gasket will in time pit out in some part of the flange nearly through the entire gasket. I have heard a great many reasons given for the cause, such as electrical action, disintegration, etc.

The wear of a gasket depends largely on the method of pulling up bolts on flanges. In fact I believe that a great many

troubles have occurred because of imperfect erection. If joints are pulled up entirely on one side and left loose on the other, and then taken up on that side, trouble with the gasket is almost certain. The bolts should be taken up gradually all around the flange. The experience of the erecting crews on high-class superheated-steam lines is an exceedingly important thing. The average steam fitter is not suited to this type of work. He has had experience with lower pressures and less important tasks, and after a piece of work is erected by him it is customary to find a great many leaks which are usually eradicated only after the whole joint has been broken and properly repaired. All these troubles can be eliminated by using only steam fitters experienced in the type of work under consideration.

Referring again to metals, cast iron and bronze, after being very severely heated and cooled off, as they naturally are in a superheated-steam line, do not seem to retain their original shape. I know of places where gate valves, after being installed in superheated-steam lines and being in service for approximately two years, are one-half inch larger face to face than when they were installed. There have also been accidents due to the fact that cast-iron flanges on wrought-iron pipe have expanded, as a total, and never returned to their former shape, thereby causing weakness and breaks.

Summing up, I believe good practice in steam lines of this type is to do away with cast iron and bronze throughout and with all fittings, using welded wrought-steel outlets, as illustrated in Figure 2, and cast-steel bodied valves with nickel-bronze seats and nickel-steel stems.

MARINE TURBINE LUBRICATION.

Read before the Institute of Marine Engineers by A. H. MATHER.

One of the results of the progress made in the design and construction of marine steam turbines, and the consequent increase in the number of vessels fitted with turbine machinery, has been the acquirement of increased knowledge as to the

problems arising in the management and running of this class of machinery. Amongst others, the problem of lubrication has forced itself upon the notice of engineers with the usual insistance which lubrication, or the lack of it, always exhibits whenever it gets the opportunity of showing itself. Marine turbines are fitted with a system of forced lubrication, which provides for a good flow of oil through the bearings, but which might be considerably improved by attention to some matters of detail considered from the standpoint of treating the oil in such a way as to ensure the best results. I have examined a number of sets of turbine machinery, and I find that while in the majority of cases every attention has been given to the mechanical details, very little thought has been given to securing the best conditions of work for the oil itself; and it has, therefore, occurred to me that a short paper dealing with this side of the subject would be both of interest and value to the members of the Institute. I do not propose to deal with the subject of turbine lubrication from any other than the mechanical point of view, and have, therefore, confined myself to the consideration of one or two matters of detail in the arrangement and fitting up of the storage tanks and pipes, attention to which would ensure that the oil in use would receive the best treatment, and would lead to a marked improvement in the running of the jobs.

STANDARD OF OIL REQUIRED.

Before entering into the consideration of these details, it might be as well just to enumerate the qualities required in an oil for this class of work:

1. It must be entirely free from saponifiable matter.
2. It must separate rapidly from any water picked up in its passage through the system.
3. It must retain its lubricating value when heated to a fairly high temperature.
4. It must retain its nature in continuous use for long periods.

In forced-lubrication systems it is essential to keep the oil

fully up to its standard, and to ensure this being done the two most important points to be considered are:

1st. Its complete separation from water picked up in its passage through the system.

2d. Its thorough filtration from grit or material substances.

FILTRATION AND SEPARATION SYSTEM.

The necessary installation on board a turbine ship consists of a tank or tanks, a filter and a force pump, and it is in regard to the arrangement of these that I propose to make a few suggestions which should result in the marked improvement in the lubrication and in economy of oil. For the purposes of this paper I have prepared a special diagram, which diagram, I may say, is based simply upon a knowledge of the properties of oil, and has been prepared with the assistance and approval of a chemist unrivalled in his intimate knowledge of oils and their proper treatment and uses. It has been recently submitted to all the prominent engineering firms throughout the Kingdom, and its principles have been adopted in a number of cases with satisfactory results.

A complete system for the filtration and separation of oil and water should consist of two storage or settling tanks, two filters, an oil cooler and a force pump.

DETAILS OF THE SCHEME.

The two tanks, which should be placed low down in the engine room to allow all the oil in the bearings and pipes to drain freely back by gravity, should each be of sufficient size to contain enough oil to fill the whole system outside of the tank, and having also an ample margin or capacity in the tank to keep a reserve of oil above the pump suction pipe. The tanks should be tilted or a pocket should be made so that the drain cock is at least two inches lower than any other part of the tank. The drain cock should be of large size, and means should also be provided to show clearly the amount of water in the tank. For this purpose a glass-sided box placed at the lower end of the tank and having adequate protection for the

glass sides is preferable to the ordinary gauge glass, as it gives a clearer and more reliable view. An air pipe should be fitted at the highest point in the tank, and led as high as necessary. The oil-return pipe from the bearings should enter the tank as near to the bottom as possible, and it is recommended that the oil be allowed to return in its heated condition in order to facilitate its separation from the water. Oil in a heated condition will also allow suspended matter to drop to the bottom more easily. In cases where a large proportion of water finds its way into the oil, as, for instance, with ordinary reciprocating marine engines fitted with forced lubrication, where it is practically impossible to prevent a considerable quantity of water from the glands, etc., falling into the crank pits and mixing with the oil, a small heater should be provided in the return pipe to raise the temperature of the oil to about 200 degrees Fahrenheit. This will result in the immediate separation of all water and foreign matter as soon as the mixture find its way into the tank, the oil rising rapidly to the top and the residue falling to the bottom. To obtain the best results this heater should be fitted to all forced-lubrication installations, including turbine sets, although no doubt it may be urged that it is adding to the cost of the plant and increasing the space occupied where the amount available is generally very limited. The pump suction should be placed as high as possible, considering the amount of oil needed in the system, so as to allow ample space for the accumulation of water and extraneous matter to remain at the bottom without being drawn through, the opening of this pipe to be directed upwards if possible. The net storage capacity of the tank is of course its capacity above this pipe opening. In the majority of installations I have examined, the inlet and outlet pipes are placed in positions the reverse of those which are here recommended, that is to say, the inlet or return pipe is led into the top of the tank, thus allowing the returning oil to fall the whole depth, thoroughly disturbing the contents and effectually preventing the separation of any of the water, and, as if this were not bad enough, the suction pipe is taken from the bot-

tom of the tank with, in some cases, an internal bend to carry it down to within an inch or so of the lowest point. This latter is done, of course, with the idea of utilizing the full capacity of the tank, but it results in keeping the turbine supplied with a mixture of oil and water. Two tanks should be supplied, and these should be used alternately at convenient intervals of perhaps two or three days, for, although the arrangements of pipes suggested allows for separation during the process of circulation, further separation is necessary to clear the oil, which can only be done by allowing it sufficient time to rest and obtain a thorough settlement.

THE FILTERS.

Two filters should be supplied to allow of one being cleaned out while the other is at work. They should consist of three or four separate layers of brass-wire gauze of, say, 24 mesh to the inch, the last or uppermost one consisting of two sheets of gauze with a sheet of cheese cloth between them. The oil should enter the filter at a point some distance above the bottom but below the layers of gauze. As the oil rises through the filter any particles of grit, dirt, etc., that may be drawn from the tank are retained in the lower part of the filter and tend to fall away from the filtering surface, which will thus remain clear for a longer period than would be the case if the oil traveled downwards, in which event all foreign matter would be retained on the filtering surface. In addition, this system of filtration would also assist in the further separation of the oil and water. For this reason sufficient capacity should be provided in the lower part of the filter, below the inlet, to collect the extraneous matter and prevent it from passing back through the pipe into the tank. A cock should be provided at the lowest point of the filter to drain this chamber.

Stop valves or cocks should be provided in the oil pipes for changing the tanks over, and it would be convenient to be able to use either filter in connection with either tank, as

one filter might, on occasion, require cleaning out when it is not desired to change the tanks.

After being filtered the oil should pass through a cooler having sufficient surface and water circulation to reduce the temperature to normal. This cooler may be of any convenient design, provided that it allows free and ample passage for the oil, and does not allow the cooling water to find its way into the oil. Probably the cheapest form to be efficient would be a series of tubes with expanded ends in a casing through which the sea water can be pumped, the tubes being arranged so that the oil passing inside them is made to travel two or three times through the water. I have seen a cooling coil fitted in the bottom of the storage tank, but this has the objection that if any leaky joints occur the water gets direct access to the oil. Another method in use is a water jacket round the tank, but this is not very efficient, as only a portion of the oil comes into contact with the cooling surface, and, in the particular instance of which I am thinking, the jacket is only supplied by a half-inch pipe, giving a very small circulation, although the tank holds probably 120 gallons. A cooler of this type should be so constructed that the cold water enters at the points where the oil leaves it before going to the bearings.

The oil pump should preferably be placed between the filter and the cooler, as there is then less likelihood of water obtaining access to the oil through leaky joints in the cooler, any leak occurring allowing a leakage of oil outwards instead of water inwards.

OTHER DETAILS.

Leaving this part of the installation, one or two matters connected with smaller details are worth referring to. The oil is distributed from the pump to each bearing, a pressure gauge being provided to show the pressure at the pump discharge. Means should be provided to show whether each separate bearing is obtaining its proper supply of oil, and also to regulate the quantity supplied to each bearing. Sight-feed

cups are fitted to a number of jobs, but some of these are so arranged that they do away with the oil pressure, the oil entering the top of a cylindrical glass cup and falling visibly into a funnel placed lower down in the cup. The pressure is, of course, lost where the oil leaves the upper pipe, and the bearing is not under forced lubrication, the pump, to all intents and purposes, simply becoming a circulating pump. A better plan is to have a similar fitting placed on the drain from the bearing which will show the amount of oil which has actually passed through the bearings while permitting the pressure to be carried right up to the working surface.

One part of the turbine which is capable of giving, and has given, very serious trouble, is the steam-packed gland where the shaft passes out of the turbine casing. The packing rings are a very close fit, and really require better lubrication than that provided by the wet steam, but the admission of oil here is a serious matter, as it simply passes straight into the turbine and through into the feed water. It seems to be a necessary precaution, however, to give the glands a little oil when working into port to prevent seizing or rusting of the shaft to the gland when the turbine is not at work, and I understand that in some cases a branch has been provided in the small pipe supplying steam to the gland to admit of a small quantity of oil being introduced when required; but a connection of this kind is a risky thing, as it would be very easy for a careless or thoughtless man to put in sufficient oil to cause boiler troubles.

In conclusion I may say that in presenting this short paper to the Institute I have done so considering that a few notes on a side issue in connection with what is probably destined to become the premier method of ship propulsion would be of general interest to all, and might be of service to some who have either the building or running of this class of machinery under their control, and if it is of use in showing where trouble may arise in the lubrication of turbine machinery, or in offering a suggestion which may lead to an improvement in the

efficiency and economy of the lubrication system generally, I shall feel satisfied that the few ideas which have suggested themselves to me in the course of my observations and experience have been worthy of placing on record.—"Page's Weekly."

SHIPS.

AUSTRO-HUNGARY.

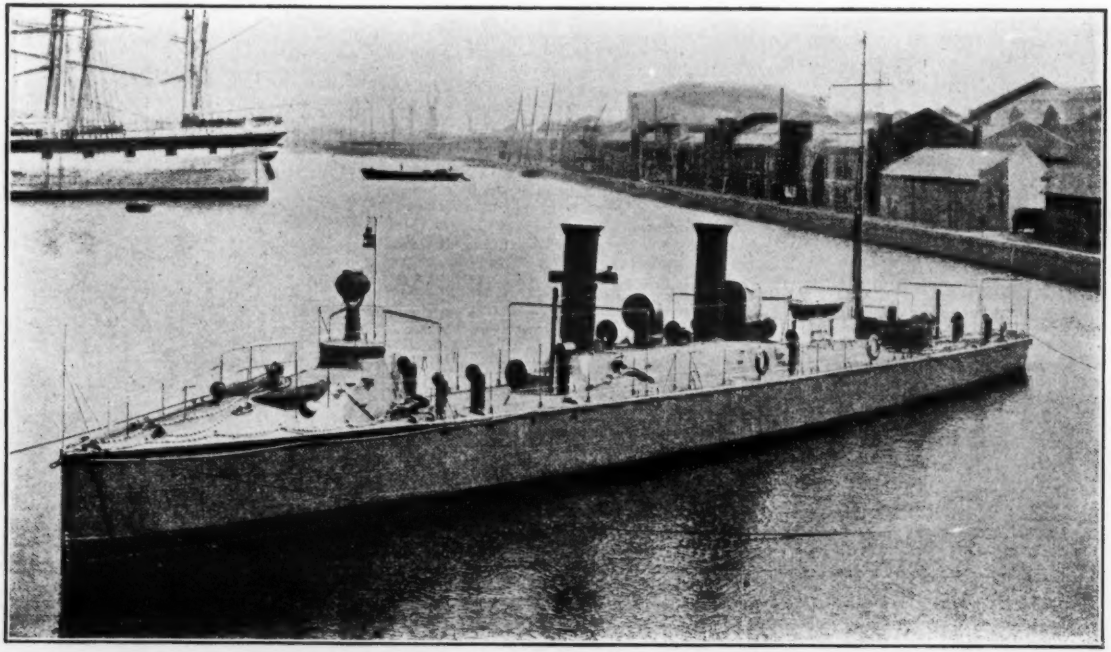
The Austro-Hungarian Government have contracted with Messrs. Yarrow & Company (Limited), of Poplar and Glasgow, for the construction of two exceptionally high-speed shallow-draught gunboats, propelled by internal-combustion engines. These vessels are to be in all essential particulars similar to *Mercury II*, which was purchased by the British Admiralty last year, and on board which, it will be remembered, the King and Queen took a trip during Cowes week.

BRAZIL.

Torpedo Boat.—We give an illustration of the first-class torpedo boat *Govaz*, which has just been built by Yarrow & Co., Limited, of London and Glasgow, for the Brazilian Government. The vessel is 152 feet 6 inches in length by 15 feet 3 inches beam, and has a displacement of about 150 tons in service condition with bunkers full. This type of topedo boat is similar to those built in recent years by Messrs. Yarrow & Co., for the Austrian, Chilian and Dutch Governments. The armament consists of two 47-mm. quick-firing guns, and two 45-cm. revolving torpedo tubes, which are to be placed on board when the vessel arrives at Rio de Janeiro.

The general internal arrangements and crew spaces are the same as is usual in this class of boat; the seamen and firemen being berthed forward, and the officers and warrant officers aft.

The machinery consists of two "Yarrow" water-tube boilers, fitted with forced draft and fed by independent feed engines. The main propelling machinery is on the Parsons' system of turbines combined with triple-expansion reciprocating engines. The reciprocating engine is used for cruising purposes, and is



FIRST-CLASS TORPEDO BOAT FOR THE BRAZILIAN GOVERNMENT.

capable of driving the vessel at a speed of 12 knots on a very low coal consumption, in which respect turbines when working far below their full power are uneconomical.

When high speeds are required the turbines are used in conjunction with this engine, which is arranged for the large variation of revolutions necessarily entailed.

The turbine machinery consists of one high-pressure and one low-pressure turbine, with the usual independent forced-lubrication pump, air pump and circulating pump.

The three-hour official full-speed trial was run on the 30th of May, in the Estuary of the Thames, in the presence of Rear-Admiral Joao Justino de Proença, Chief of the Brazilian Naval Commission, Captain Josè Thomaz Machado Portella, Chief of Naval Construction, and Captain Bartholomeo Francisco de Souza e Silva, Chief of the Machinery Section; and during this trial six runs were made over the Admiralty measured mile on the Maplin Sands, with the following results:

Steam pressure in boilers, 230 pounds per square inch; in cruising engine, 207 pounds per square inch; at H.P. turbine, 220 pounds per square inch; at L.P. turbine, 29.6 pounds per square inch; vacuum in condenser, 27 inches; air pressure in stokehold, 1.8 inches.

For the three hours the mean speed was 26.493 knots.

Following the above, a consumption trial was made in the presence of Captain Portella and Captain Bartholomeo da Silva, when a mean speed of 11.277 knots per hour was obtained, with 150 pounds steam pressure in the cruising engine, the turbines being shut off, and the coal consumption under these conditions showed 56 knots run per ton of coal burned.

The above figures fully justify Messrs. Yarrow's decision in adopting the combination of reciprocating engines and turbines first suggested to them by Captain Soliani, of Genoa, in view of the fact that turbines have not yet been found economical at low power, even when a special cruising turbine has been fitted. Another difficulty avoided by this system is that of going astern, the reciprocating engine being fitted with reversing gear, as usual, and no astern turbine is, in consequence, nec-

essary. It may, therefore, be said that the cruising engine takes the place of a cruising turbine and an astern turbine, and, of course, weighs considerably less, and in addition has the great advantage of being an engine the design of which is familiar to every engineer. No difficulty is met with in arranging the reciprocating engine to run at the revolutions necessary for full speed, as the slip of its propeller reduces as the speed of the vessel increases, so that although the speed of the boat rises, say, from 11 to 26 knots, the speed of the reciprocating engine only rises from about 340 revolutions to 540 revolutions per minute. Hence, though the speed is considerably more than doubled, the revolutions of the cruising engine only increase some 60 per cent.—"Engineering."

Customs Cruiser for Brazil.—A twin-screw cruiser yacht named the *Amapa* has recently been built by Messrs. John I. Thornycroft & Co., Limited, at Woolston Works, Southampton, for the Customs service of the Brazilian Government. She is constructed of steel to Lloyds "100 A. I." class, and on the official trial, which was carried out on May 28th last, proved herself, so we are informed, an excellent seaboat, and maintained a speed of 12½ miles, with natural draft. This was, it may be added, half a mile in excess of the contract speed. A speed of 15¾ miles was attained with forced draft, this being three-quarters of a mile above the contract speed. Her principal dimensions are: Length between perpendiculars, 130 feet; beam, molded, 17 feet; depth, molded to main deck, 9 feet 9 inches. A chart house with a flying bridge over it is placed at the aft end of forecastle deck, on which are situated the searchlight, steam-steering gear, compass and telegraph. A three-pounder quick-firing gun is mounted on the forecastle head, and a combined steam and hand windlass is also provided. The accommodation for the officers and guests is under the raised quarter deck aft, while the petty officers and crew are berthed forward. The galley is on the main deck. Stores, fresh-water tanks, magazines and ballast tanks are arranged under the cabin flats.

The vessel is lighted throughout by electricity, and fresh air

is circulated through the cabins by means of electric centrifugal fans. Three boats are carried. One of these is a 25-foot lifeboat, another a 25-foot cutter, and the third a 12-foot dinghy. Two Downton pumps are fitted for clearing the bilges and for washing deck and fire purposes.

The machinery consists of two sets of triple-compound surface-condensing engines working at 300 revolutions per minute, having cylinders of 9 inches, 13 inches and 20½ inches diameter, and 11 inches stroke. Steam is supplied from a water-tube boiler working under forced draft on the closed-stokehold principle, the working steam pressure being 180 pounds per square inch.

ENGLAND.

The New Ocean-Going Destroyers.—The first five ocean-going destroyers of the Tribal class—so appropriately named *Afridi, Cossack, Ghurka, Mohawk* and *Tartar,* respectively—are approaching completion.

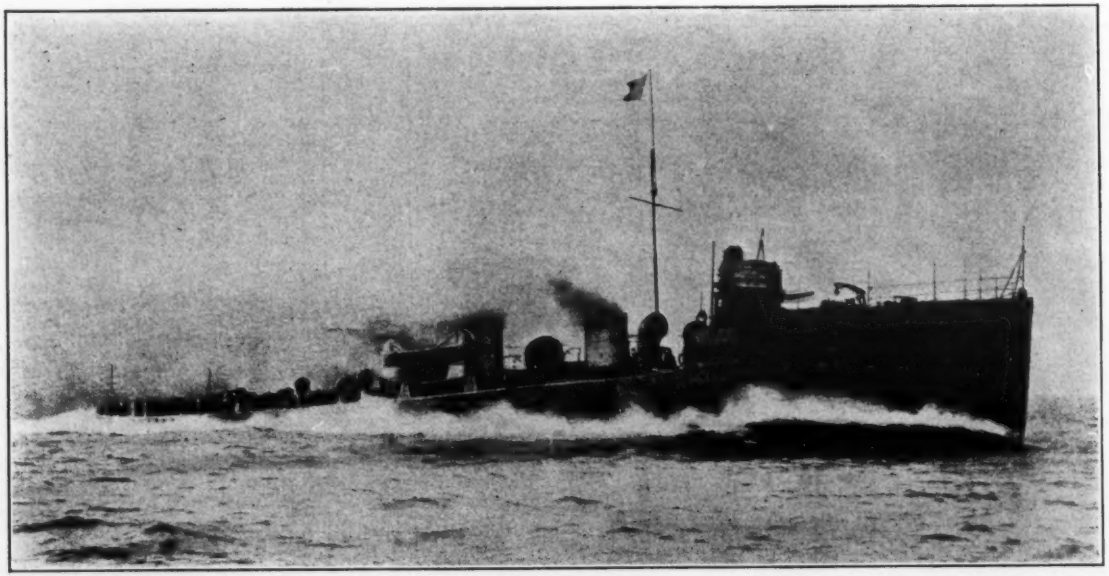
The *Cossack* is 270 feet in length, 26 feet in width, and has a depth in hold of 15 feet 5 inches. She is fitted with Parsons turbine engines, which have been constructed by the builders at their Birkenhead Works. They are arranged on three shafts, the high-pressure working the center screw, and the low-pressure and cruising turbines driving the wing propellers. Steam is supplied by five water-tube boilers of the Express straight-tube type, working at a pressure of 220 pounds per square inch, and fired with oil fuel by sprayers of the Admiralty type. Her armament consists of three 12-pounder quick-firing guns, one on each side of the forecastle, and one on a raised platform aft. There are also two deck torpedo tubes.

Very special care was taken in the design and arrangement of the engine room to ensure that the ventilation should be as ample and efficient as possible, considering the inherent tendency of turbine spaces to be excessively hot. The after half of the aftermost funnel casing does duty as a huge up-cast immediately over the hottest part of the engine room—viz: the main steam pipes and their stop and regulating valves. In

addition, two ventilating fans are fitted to induce currents of air to keep moving in the required direction. The level of the starting-platform is kept low, and is so arranged that the engineer in charge has full control of the starting gear, the temperature at this point being quite normal. The extremely neat and compact arrangement of starting and maneuvering gear is worth noting. The large valves are all of Cockburn's patent flexible-disc type, and are easily operated by hand. The ahead and astern maneuvering valves for starboard and port respectively are each operated by a single hand wheel. The regulating valves to the cruising turbines are fitted on branches cast on the main regulating-valve box, and are placed underneath the maneuvering valves, one on each side.

The trials for this class of vessel are unusually exacting and exhaustive, and her builders, Messrs. Cammell, Laird & Co., Limited, are to be congratulated on the fact that their vessel is the first of her class to complete all the trials required by the Admiralty prior to fitting out for final completion ready for commissioning. For the high-speed trials it was decided to take the vessel to the Clyde, whither she proceeded at the end of September, and, on arrival there, a preliminary trial on the measured mile was carried out, and was succeeded by an official one-hour establishing trial at full speed, to determine whether the vessel and her machinery were in a sufficiently satisfactory state to make the full-speed six-hours' continuous-steaming trial, which, undoubtedly, is the most severe, as well as the most interesting, of the official trials that vessels of this type have to carry out.

The conditions under which this trial is made are of such a nature as to make it possibly disappointing to those who are in charge, owing to the fact that for the first three hours the vessel's speed is calculated from the number of revolutions obtained during that period, without the exact knowledge of what the actual speed may prove to be as the result of running the miles during the fourth hour; and from this cause a curious situation arose, as it was demonstrated on the fourth hour that the speed obtained was not equal to what had been antici-



H. M. Destroyer "Ghurka."

pated by previous trial runs. In order, therefore, to obtain the speed of 33 knots over the complete period of six hours, special efforts had to be made to increase the revolutions, and consequently the speed, during the final two hours; and despite these efforts, the result, unfortunately, was that the average speed of 33 knots for the six hours was not quite realized, the actual speed being 32.977 knots, or a deficiency of 0.14 of a knot, or less than 300 yards in a total of 198 nautical miles, equivalent in time to 15 seconds.

Notwithstanding this very close result, the firm decided to make another trial, which was run a few days later, and the vessel was successful on that occasion in obtaining a speed of 33.1 knots, equivalent to 198.6 miles, during the six hours.

This mileage run during the six hours constitutes a record, inasmuch as no other vessel, we believe, has ever previously covered such a distance in the time named, although, no doubt, with the experience gained the result may be exceeded by some of the other firms who are building sister vessels.

We give below the actual speeds obtained in each individual hour in both of the above-mentioned trials:

	First trial. *Knots.*	Second trial. *Knots.*
Average speed for first hour	32.58	32.58
second hour	32.66	32.78
third hour	32.85	33.34
fourth hour	32.71	32.85
fifth hour	33.47	33.31
sixth hour	33.59	33.64
Mean speed	32.977	33.1

It must be borne in mind that at this extremely high speed the vessel is rapidly lightening as the trial proceeds, which accounts for the varying speed, although the speed during the fourth hour, during which the runs on the measured mile are made, is somewhat interfered with by the turnings at the end of each run when on the mile.

Shortly after the completion of the speed trial other subsidiary trials were carried out, and occasion was taken, during the vessel's return to Birkenhead from the Clyde, to run the

stipulated 24-hours' oil-fuel consumption trial at the economical speed. This trial, despite the fact that it was carried out in somewhat unfavorable weather, was entirely satisfactory, and dispelled a doubt which had existed in the minds of the builders as to what the result would be with this type of machinery, and the conditions under which the trial had to be made.

The contract radius of action of 1,500 knots was exceeded, the distance run being, on the stipulated consumption of oil, 1,580 knots, the other requirements being complied with in all respects.

The following are further interesting particulars of the vessel's performance:

Mean speed on the measured mile, 33.15 knots; highest speed on an individual run on the mile, 33.65 knots; consumption of oil for 24 hours, 27 tons; distance run per ton of fuel at the economical speed, 12.5 nautical miles.

Subsequent to the vessel's return to Birkenhead other important specified trials have been made, among which mention may be made of the gun-firing and torpedo trials, astern, circle, steering and anchor trials. On the occasion of the torpedo trials opportunity was taken to fire torpedoes whilst the vessel was steaming at full speed, and we understand the results that were obtained in this direction were eminently satisfactory and constituted a record as regards discharging at this high rate of speed.

Throughout the whole series of trials, including progressive trials, which were carried out by the builders, shaft horsepowers have been taken by "flash-light" torsion meters, supplied by Chadburns, Limited, of Liverpool. The results were checked and counter checked by turbine pressure and steam consumption in every possible way during the early trials, until, finally, the powers obtained were accepted with full confidence that the percentage error was less than obtains with ordinary steam-indicator diagrams. It has been of great interest to observe the variation in the distribution of power relatively through the three shafts, under different conditions of working the turbines, such as cruising alternately high and intermedi-

ate, and with slight overload. The shaft horsepower was easily read down to such low powers as 820, which was the power required at a speed of 13 knots, the lowest speed taken during the progressive trials. The facility of the system employed, including power diagrams for each shaft, enabled the shaft horsepower to be read off immediately on the completion of any interval of time during which the revolutions had been taken or at the end of each mile.

The vessel has proved herself throughout all the trials a thoroughly substantial sea-going vessel, absolutely free from vibration, and altogether one of the most interesting units that will be added to His Majesty's Fleet. She will now be pushed forward as rapidly as possible for completion and commission, and, we understand, will probably be delivered before the end of the present year.

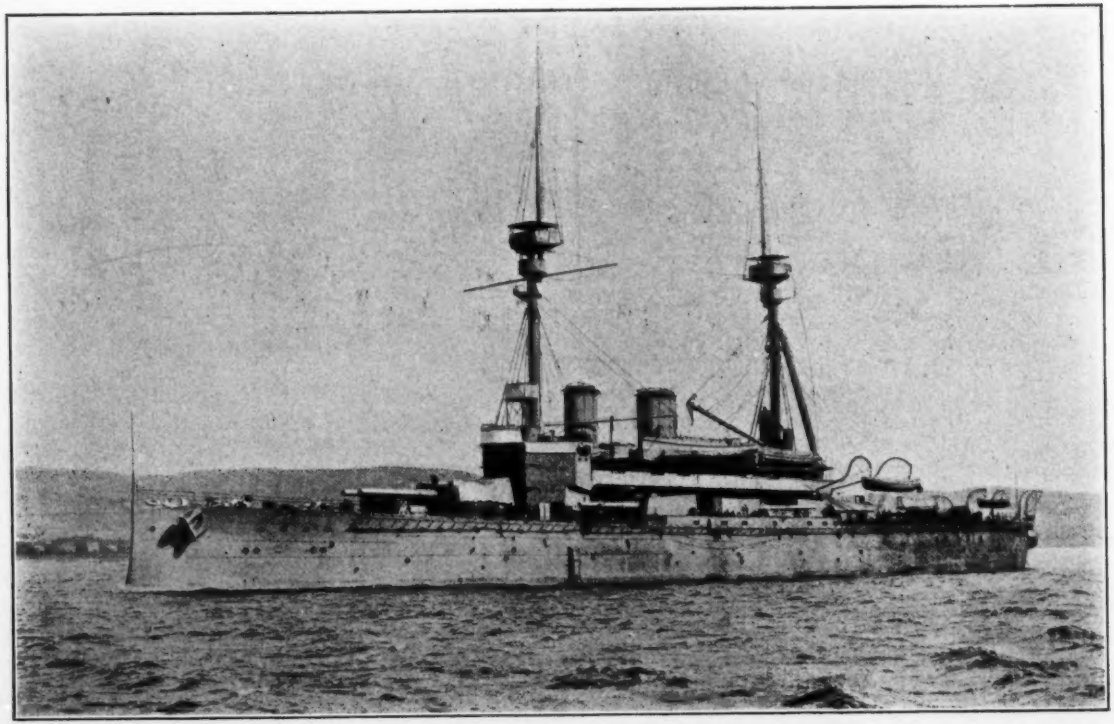
Messrs. Cammell, Laird & Co. have reason to be satisfied with the success of the *Cossack,* coming as it does in sequence to the various sizes and characteristics of destroyers since the date of their inception. This firm built five of the earliest of the 27-knot destroyers some thirteen years ago, and all of their vessels—the *Ferret, Lynx, Banshee, Contest* and *Dragon*—are still, we believe, in commission. These vessels were not equal in size, though termed destroyers, to the now up-to-date first-class torpedo boats.

Messrs. Cammell, Laird & Co.'s record through the various stages of the development of destroyers has been remarkable, they having built probably more of this class of vessel than any other firm in existence. No less than thirteen of the 30-knot destroyers have been built by them, the earliest of which was the *Quail,* launched in 1895, and the latest, the *Lively* and *Sprightly,* in 1900. The *Orwell,* one of the 30-knotters, it will be remembered, was cut in two in the Mediterranean, completely losing her fore end, and proved herself sufficiently seaworthy in that condition to reach Malta under her own steam, at which station her repairs were carried out.

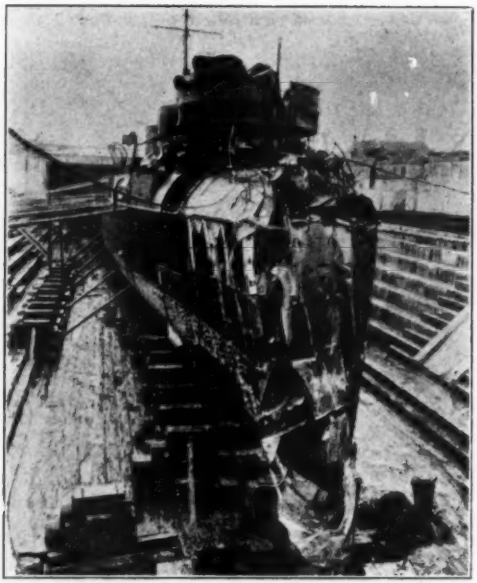
They have also constructed five destroyers of the River class,

a heavier and more seaworthy class of vessel, of 25½ knots speed, all of which have proved entirely satisfactory. They, however, have still to face a far more difficult problem than the *Cossack* in bringing to a successful issue the construction and trials of H.M.S. *Swift,* a vessel of about double the power of the *Cossack,* and a displacement of about 1,800 tons, and designed for a speed of 36 knots. This remarkable development of the torpedo-boat destroyer, starting from a displacement of 325 tons and attaining in the course of thirteen years to 1,800 tons, constitutes in itself a record of wonderful progress.—"Engineering."

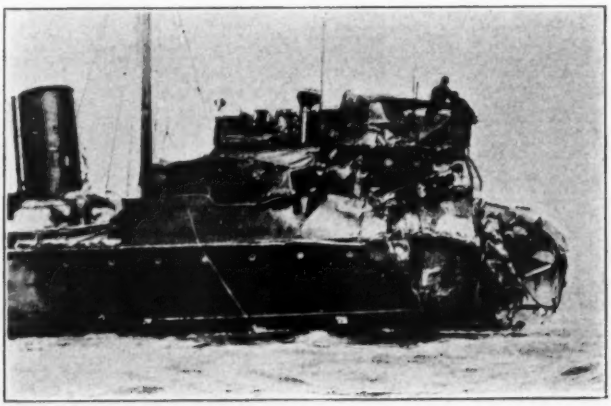
H. M. Battleship Agamemnon.—The new battleship *Agamemnon* attained on her speed trials a speed of 18.752 knots when the engines were developing 17,285 indicated horsepower, whereas the contract anticipated 18 knots when the machinery indicated 16,750 indicated horsepower. The *Agamemnon* is thus about equal in speed with the vessels of the *King Edward VII* class, and much superior to them in gunpower. As shown in the illustration, which is reproduced from a copyright photograph by Messrs. Maclure & Macdonald, of Glasgow, she mounts four 12-inch and ten 9.2-inch guns, instead of four 12-inch, four 9.2-inch, and ten 6-inch guns in the earlier vessels. There are many who prefer the *Agamemnon,* so far as armament is concerned, to the *Dreadnought* class, and two of the later ships built in Japan are equipped with two calibers, like the *Agamemnon* and *Lord Nelson,* while the French in their latest class are also adopting a combination of gun corresponding to that in the *Agamemnon.* This preference is due to the greater rapidity of fire of the lesser-caliber gun, but it must always be remembered that although these give a greater volume of shot per unit of time, their range for all of the guns is not so great. On the basis that the 9.2-inch gun may fire four rounds per minute as compared with two rounds of the weapon of 12-inch caliber, and that the 9.2-inch gun uses a projectile of 380 pounds, and the 12-inch gun a shot of 850 pounds, we have the following comparison of the broadside of the two ships:



H. M. Battleship "Agamemnon."



THE "KESTREL" AFTER COLLISION WITH THE "TEVIOT."



THE "QUAIL" AFTER COLLISION WITH THE "ATTENTIVE."

		Pounds.
12-inch	4 × 850 × 2 rounds per minute	= 6,800
9.2-inch	5 × 380 × 4 rounds per minute	= 7,600
Total per minute		= 14,400

and for the new *Dreadnought* class:

12-inch	8 × 850 × 2 rounds per minute	= 13,600
Difference		800

With regard to direct bow or stern fire the superiority lies with the *Dreadnoughts:*

Dreadnoughts:		
12-inch	6 × 850 × 2 rounds per minute	= 10,200
Agamemnon:		
12-inch	2 × 850 × 2 rounds per minute	= 3,400
9.2-inch	4 × 380 × 4 rounds per minute	= 6,080
		9,480
Difference		720

Of course the energy of the 9.2-inch guns is considerably less than that of the 12-inch; the former is capable of piercing 13½ inches of hard-steel plate at a range of 3,000 yards, against 19½ inches in the case of the 12-inch gun. The height of the center of the two forward 12-inch guns is 27 feet, and the two after 22 feet above the water line, and the average height of the centers of the 9.2-inch guns is 23 feet above the water line. The 12-pounder (18-cwt.) guns of the latest pattern are placed on the flying deck, a superstructure about 13 feet above the upper deck, extending from the forward to the after 12-inch gun barbette, and about 40 feet in width. The centers of these guns are therefore about 34 feet above the water line, and they are thus admirably situated in commanding positions for repelling torpedo-boat attack. The *Agamemnon* was built by Messrs. William Beardmore & Co. at their Naval Construction Works, Dalmuir, and the machinery was supplied by Messrs. Hawthorn, Leslie & Co., Limited, of Newcastle-on-Tyne.

Accidents to British Torpedo-Boat Destroyers.—There has been a regrettable series of accidents to torpedo-boat destroyers of His Majesty's Navy. With the exception, how-

ever, of the fire on board the destroyer *Spiteful,* the accidents were not attended with fatal results, although in one case some six or seven men were injured.

This latter accident occurred to the destroyer *Quail,* off the Isle of Wight. The *Quail,* one of the defending force, was engaged in assisting to repel an attack on Poole Harbour by the scouts *Attentive* and *Adventure.* The *Attentive,* traveling, it is reported, at about 20 knots, struck the *Quail* at right angles, cutting about 40 feet of the bows clean away. Being in action all water-tight compartments were closed at the time of collision, and the vessel remaining afloat was towed stern first to Portsmouth. The portion cut off by the *Attentive* sank, and several of the crew had rather narrow escapes, but no lives were lost.

Another accident occurred off Portland, to the destroyers *Teviot* and *Kestrel.* The vessels were both traveling at full speed at the time, with lights out, and the *Teviot* cut into the *Kestrel* for a length of about 30 feet. The damaged plating, etc., on the starboard side broke off and sank, while that on the port side was bent round, covering to some extent the gap on the starboard side. The *Teviot* did not suffer to the same extent, although her bows were buckled and damaged.

New Dreadnoughts.—Although the dockyard authorities at Portsmouth and Devonport are very reticent on the subject of the new *Dreadnoughts,* for which the building designs have already been received, there is reason to believe that the armament of these battleships will show no departure from the original *Dreadnought* specification, saving that the manner of disposition will be different, and that the anti-torpedo battery will consist of 4.7-inch guns instead of 12-pounders. It had been proposed to equip the new warships with 13.5-inch guns of an entirely new pattern, and several of these weapons have actually been built up. They weigh about 80 tons without mountings, and fire a projectile of 850 pounds. The idea had been to mount eight of these in each ship of the new group, instead of ten 12-inch guns, as in the original *Dreadnought.* But, so far as can be gathered, this project has now been definitely aban-

doned. The modified plan of distribution of armament in these latest battleships is the outcome of the discovery of imperfections in this way in the *Dreadnought.* At certain angles in the radius of her fire delivery several of the guns of this ship are masked by the turrets of other guns. Moreover, she could never discharge more than eight of her ten pieces in one broadside. In the new vessels the turrets are to be placed in a center line, so that the whole of the guns can be carried on either broadside. To obviate masking in either astern or ahead fire, the gun positions will be successively raised upon solid platforms, so as to give the rear weapons a clear range over the tops of the intervening turrets. The statement that it was proposed to place three guns in each turret is incorrect. Only two guns will be mounted in each of these citadels.

A 20,000-ton battleship is to be laid down immediately at the Devonport dockyard for the English Navy, being the fifth battleship of the *Dreadnought* class.

Temeraire, H.M.S., new battleship; 490 by 82 feet. There are now three ships of the *Dreadnought* class afloat—one in commission and two in course of completion. A fourth, the *Superb,* is building at Elswick by Sir W. G. Armstrong, Whitworth & Co. In design the *Temeraire* is like the *Dreadnought;* but, as in the case of the *Bellerophon,* recently launched at Portsmouth, some improvements will be introduced as the result of the experience gained during the trials of the *Dreadnought.* The vessel has a displacement of 18,400 tons. With turbine engines of 23,000 H.P. she will make a speed of 21 knots. As in the other ships of the class, the main armament will consist of ten 12-inch breech-loading guns, but it is understood that 4-inch guns will be mounted in place of the 12-pounders which the *Dreadnought* carries to repel torpedo attacks. The main defensive armor will be a belt of 12-inch hardened steel amidships, but there is likely to be a slight departure in the protective armament of other parts of the vessel. Among other improvements to be introduced is steering gear of a new description. The *Temeraire* was laid down on 1st January last, and is expected to be completed within two years

of that date. In the *Temeraire* and her two sister ships, the *Bellerophon* and *Superb,* a rearrangement of the 12-inch guns is introduced, the effect of which is to enable eight of these pieces to be fired astern as well as broadside. This has been done by raising the turret amidships on the quarter deck so as to permit the two guns which it contains to be fired over the after turret, which is also in the center line. The substitution of 4-inch guns for 12-pounders necessitates a rearrangement of the battery, and this, together with the alteration in respect of the big guns, has somewhat added to the displacement of the ship, which is 700 tons greater than that of the *Dreadnought.* Although the provision for the protection of the ships of this class against attacks by torpedoes or mines remains secret, there is reason to believe that in the last three vessels of the class not only have the armor arrangements been varied, but the cellular construction of the hull has been modified to provide a means of comparatively harmless exit for the gases set up by an under-water explosion. The whole of the machinery for this vessel is being constructed by Messrs. R. & W. Hawthorn, Leslie & Co., the turbine propelling machinery being of the Parsons type, driving four screws and giving 23,000 I.H.P., corresponding to a speed of vessel of 21 knots. —"The Steamship."

The new British 33-knot Torpedo-boat Destroyer Tartar. —The *Tartar*, one of the new class of 33-knot turbine destroyers, recently completed for the British Navy by Messrs. John I. Thornycroft & Co., Limited, of Southampton, ran a very successful contractors' preliminary trial in the Solent over the British Admiralty measured course on the 20th November, when the mean speed obtained on six runs was 33.122 knots, or 38.1 English land miles, without any special effort.

The vessel proved itself entirely free from vibration even at the highest speed, and in this feature stands alone among torpedo craft, though the introduction of the turbine has done a great deal to lessen the vibration usually associated with similar vessels using reciprocating engines.

The power is produced by turbines built by Messrs. Thorny-

croft, arranged on three shafts and developing approximately 14,000 I.H.P. together. The H.P. turbine on center shaft exhausts into the L.P. turbines on the wing shafts. Forward of the L.P. turbines is a H.P. and L.P. cruising turbine, each driving a shaft with one propeller. Steam is supplied by six water-tube boilers of the latest Thornycroft type, which use oil fuel exclusively, and work at 220 pounds per square inch. The fuel carried is sufficient for between 1,500 and 1,700 nautical miles.

The *Tartar* is 270 feet in length on water line, 26 feet in width and 17 feet 2 inches in depth. Her shape follows generally the lines of the earlier destroyers by the same firm, a prominent feature of which is the graceful turtle deck, which takes the place of the raised forecastle deck generally adopted for vessels of this size. The advantage of the former is that the water is thrown overboard instead of under the upper deck.

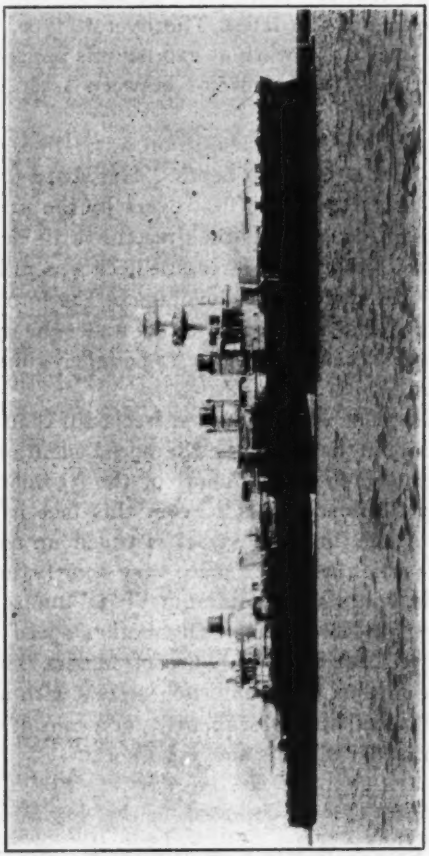
The armament consists of three 12-pounder guns and two torpedo tubes.

The vessel is of greater beam and freeboard than others of the same class, but nevertheless the speed mentioned above is the greatest yet obtained by any of the 33-knot boats on contractors' preliminary trials. From this fact it is safe to assume that by the time the vessel is tuned up ready for its official trials (which will take place very shortly, and consist in running six hours at a minimum of 33 knots per hour) there is every likelihood that still another speed record for torpedo craft will fall into the hands of Messrs. Thornycroft, who were the first to construct such vessels with a speed of more than 28 knots, and subsequently, for a considerable time, held the record for 30 knots and 31½ knots.

FRANCE.

Trials of French Battleships Democratie and Justice.—These two battleships, which have recently finished their official trials, are the third and fourth of the 1900 Program, being sisters of the *République* and *Patrie* and of the *Liberté*, and *Vérité*, which are nearly completed. The ships have a

load water-line length of 439 feet, an extreme beam of 79 feet 7 inches and a draught of 27 feet 5 inches. The displacement is 14,867 tons, with a block coefficient of 0.543. An armor belt, with a maximum width of 18 feet 5 inches, has



THE FRENCH BATTLESHIP "JUSTICE" AT ANCHOR.

an extreme thickness of 11 inches, tapering to 3 inches at the ends. There are two protective decks.

The battery, which is a powerful one, consists in the first two of the type of four 12-inch guns mounted in pairs in two

turrets, and eighteen 6.4-inch rapid-fire guns, of which twelve are in pairs in turrets and six are in casemates. The secondary battery includes twenty-seven 3-pounder and 1-pounder guns, and five torpedo tubes, two of which are submerged. A considerable difference exists in the battery of the last four ships of the type, through the substitution of ten 7.6-inch guns and eight 3.9-inch guns for the eighteen 6.4-inch guns on the first two vessels. This change makes for increased strength of broadside, particularly at long ranges. The 7.6-inch guns are located, six each in a single turret and four in casemates.

Each of the ships is propelled by three screws operated by triple-expansion four-cylinder engines in separate watertight compartments. The cylinder diameters are 35, 49, 55 and 55 inches, with a stroke of 41½ inches. Steam is supplied by twenty-four water-tube boilers, located in three boiler rooms, each with its own funnel. The total grate surface is 128 square meters (1,378 square feet), with a heating surface of about 46,000 square feet, making a ratio of 33.4 to 1. The boiler pressure is 251 pounds per square inch. The normal coal supply of 900 tons gives an estimated steaming radius at ten knots of 4,200 nautical miles. With the full coal supply of 1,850 tons the estimated radius is increased to 8,500 miles.

It must be pointed out that for the first time since the days of the old sailing fleet France possesses a homogeneous squadron of six vessels of high power, it having until recently been the French naval policy to construct contemporaneous vessels on the divergent designs of various naval architects, resulting in the acquisition of a fleet which was described by one of the French admirals as a "museum."

The first two ships of this type were compared in our August number with the battleship *Virginia*, of the United States Navy. The last four ships have a more powerful battery than the first two, but even with this advantage they could scarcely be expected to compete successfully with the ships of the *Virginia* class under ordinary conditions.

The results of the trials of the *Démocratie* and *Justice* are

given in the table. *Démocratie* is fitted with Belleville boilers, while the *Justice* has Niclausse generators. The poor propulsive efficiency at low speeds is more than offset by the magnificent results at 17 to 18 knots, while the figures for full speed are extremely good. It will be noted that in passing from the intermediate to the highest speed the index n in the expression $H_1 : H_2 : : V_1^n : V_2^n$ is very high, indicating a limit in highest propulsive efficiency at about 18 knots. This index, which is often assumed as 3, becomes 4.58 for the *Démocratie* and 5.98 for the *Justice*.

Six-hour trial at low power.	Contract.	*Démocratie.*	*Justice.*
Boilers at work	5	5	5
Grate surface, square feet	287	287	287
Indicated horsepower	2,500	2,584	2,541
Consumption of coal, in pounds :			
Per horsepower hour	...	1.234	...
Per square foot of grate	...	...	15.2
Mean speed in knots	...	7.5	7.8
Admiralty constant	...	98.8	112.8
Normal steaming radius	...	4,665	...
Twenty-four-hour consumption trial.			
Boilers at work	...	24	22
Grate surface, square feet	...	1,378	1,262
Indicated horsepower	10,500	11,472	11,520
Consumption of coal, in pounds :			
Per horsepower hour	...	1.484	1.528
Per square foot of grate	...	...	14.8
Average speed in knots	...	17.39	17.94
Admiralty constant	...	277.6	303.2
Normal steaming radius	...	2,027	2,023
Full-power three-hour trial.			
Boilers at work	24	24	24
Indicated horsepower	17,500	19,190	18,548
Consumption of coal, in pounds :			
Per horsepower hour	...	...	1.824
Per square foot of grate	24.3	24.3	23.7
Average speed in knots	18	19.44	19.43
Admiralty constant	201.6	231.6	239.1
Normal steaming radius	...	...	1,140

—"International Marine Engineering."

Latest French Submarine Boat.—The latest and largest type of French submarine boat, which is shortly to take its place in the Channel Defence Squadon, is illustrated in the engraving given in Fig. 3. She is named the *Emeraude,* and was ordered on the 24th October, 1903, and was laid down on December 1, 1904. She was launched on the 6th of August of last year, since when an extensive series of trials and experiments have been carried out, which are now completed. Her chief dimensions are: Length over all, 146 feet 6 inches; extreme breadth, 12 feet 10 inches; depth, 12 feet 1 inch. For ordinary navigation she is provided with two sets of internal-combustion engines, which together develop 600 horsepower. Each engine drives a propeller shaft. For submarine navigation a couple of motors are used. On trial a speed of 12 knots when three-quarters afloat, and 8 to 9 knots when half afloat, was attained. The *Emeraude* has a peculiar appearance, her hull being similar to the body of a large whale. The little craft, it is said, dives splendidly, but when afloat in rough weather she does not navigate so well as the "submersibles." For the officers and crew the accommodation is more comfortable than in the older type of boats, and, as will be seen from the engraving, the navigating bridge is large, and the visual radius for searching for an enemy is largely increased. Each of the three officers which are carried has his own room, and there are nineteen men in the crew. The armament is composed of six 18-inch torpedoes. The *Emeraude,* being the first of her class, has been subjected to very severe trials, but the five others, one of which has already been launched, will be completed in much less time. We are informed that the cost of building and equipment came to about £63,668.

In order that a comparison may be drawn between the *Emeraude* and what was hitherto the latest French submarine, we give an illustration in Fig. 2 of the "Y" type of boat. The Y has a length of 142 feet 2 inches, beam 9 feet 10 inches, and a draught of 7 feet 8 inches. It has a displacement of 213 tons and a horsepower of 250; her maximum speed above water being 11 knots, and when submerged 8 knots. She has two tor-

pedo tubes and carries a complement of fifteen officers and men. Fig. 1 shows the French submersible *Sirene,* which was laid down in 1900. She is 110 feet 6 inches in length, 11 feet 3 inches beam, and has a draught of 5 feet 2 inches. Her engines develop 217 horsepower and drive one propeller only. On trial she attained a speed of 12 knots. The armament comprises four torpedo tubes, and she is manned by two officers and eight men.—"The Engineer."

GERMANY.

German Torpedo Boat Construction.—The steady addition of torpedo craft of large size to the Imperial German Navy continues unchecked, and their value for war purposes is somewhat disguised by the official system of frequently describing them as torpedo boats, instead of classing them, as would be the case in England, as "destroyers." In general, their torpedo equipment is greater than that of British vessels of corresponding date, while their guns are fewer and less powerful; their hulls are of relatively heavier scantlings, and the vessels themselves have proved to be excellent sea boats, though slightly inferior in trial-trip speed to those of other nations.

No less than twenty-two boats of about 420 tons service displacement—*G 110* to *G 113* and *S 114* to *S 131*—were constructed by Messrs. Krupp, at the Germania Works, and by Schichau, at Elbing, between 1902 and 1905. Differing only slightly in detail, they are all about 205 feet long by 23 feet beam, and have bunker capacity for 100 tons of coal; at 15 knots the service radius of action is about 1,400 miles. Their trials were run at displacements varying from 380 to 415 tons, and speeds between 27 and 28 knots were obtained with about 6,000 to 6,500 indicated horsepower. *S 125* of the latter division, built by Schichau, was the first vessel in the German Navy fitted with Parsons' turbines. On her official trials she attained 26.7 knots at 440 tons, the increase in displacement being due to the greater amount of coal on board compared with her sister ships. The armament consists of three 4-pounder quick-firing guns, two machine guns, and three 450-mm. torpedo

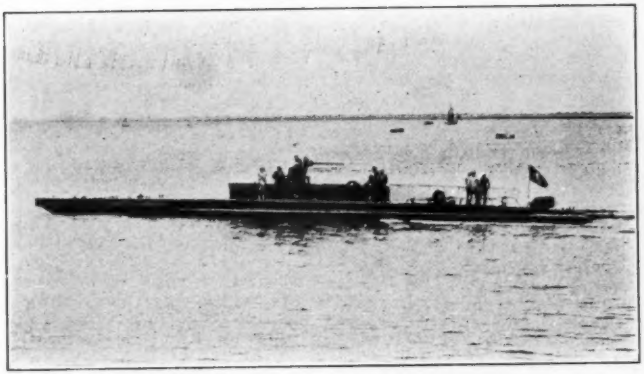


Fig. 1.—French Submersible "Sirene."

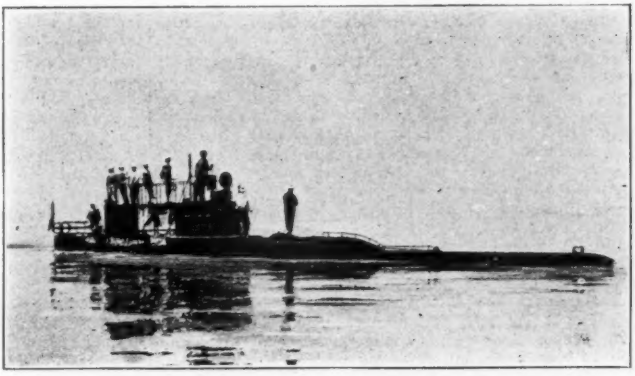


Fig. 2.—French Submarine of the "Y" Class.

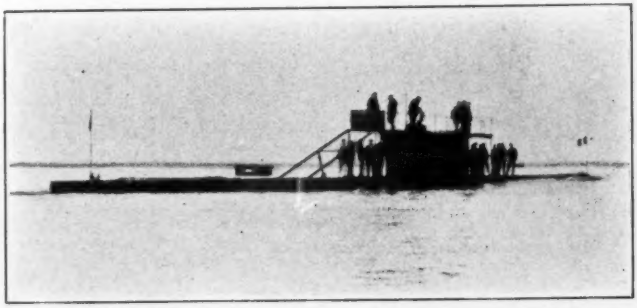
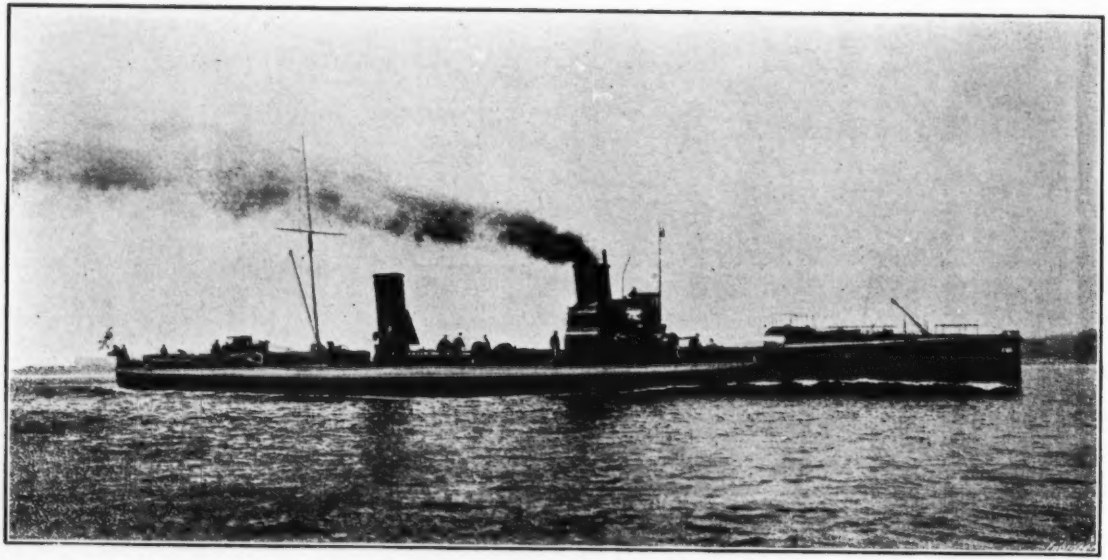


Fig. 3.—French Submarine Boat "Emeraude."



GERMAN TURBINE-PROPELLED DESTROYER "S 125."

tubes. Comparing these with British practice, we find that during the same period thirty-five "River" class destroyers of 570 tons were added to the Royal Navy, each carrying one 12-pounder and five 6-pounder quick-firing guns, but only two tubes. Their average full speed was barely 26.0 knots, but on their greater dimensions—225 feet in length by 23 feet 6 inches beam—they carry 125 tons of coal, and their radius at 15 knots is nearly 1,800 miles.

The next batch of German boats—*G 132* to *G 136*—was ordered in 1905, and consisted of 460-ton boats of 28 knots speed, carrying, as usual, three torpedo tubes, as well as four 5-pounder quick-firing guns and two machine guns. All these vessels were launched in 1906, and are now either in service or being completed at the Germania Works. They are almost identical with the previous destroyers, and, being built by only two firms, present fewer divergencies of appearance than is the case with the various British boats—Laird's and Palmer's, for example.

Since the commencement of 1907 several 30-knot boats, ordered in 1906, have been launched or have run their trials. *G 137* really belongs to this class, though she was ordered in December, 1905; the others consist of the Schichau division, *S 138* to *S 149*. These vessels all carry one 15-pounder, three 5-pounders, two machine guns, and three torpedo tubes. The designed displacement was 570 tons in the case of *G 137*, and 530 for the other boats; the indicated horsepower was to be 10,000. Four Schulz boilers are fitted, as against three in the earlier boats, the pressure being 230 pounds per square inch. In point of size they closely approach the dimensions of the "River" class, but are about 10 feet longer and 1 foot wider. About eight of the "S" class have run their trials, and, at a reduced displacement, have only attained 29.6 knots, though the horsepowers ran up to nearly 11,000. Considerable difficulty was involved in reaching even this speed, and numerous propellers were tried without success. The remainder of this batch is now approaching completion.

The latest destroyer division to be built for the German

Navy was ordered early this year from the Vulkan Company, which was entrusted with torpedo-boat work for the first time, and consists of twelve 580-ton boats *V 150* to *V 161*. Like *G 137* they are 235 feet long by 24.10 beam, and carry the same armament. As in most of the German destroyers, the engines are placed in two compartments, one abaft the other, and the coal bunkers, of 115 tons capacity, are extended at the sides of the engine room for purposes of protection. The designed speed is 30 knots, and in view of the performance of *G 137*, it will be interesting to see how these vessels compare with her. At present they are approaching the launching stage.

G 137 is fitted with marine turbines of the Parsons type, and on her recent official trials on the German Admiralty mile at Neukrug, attained a maximum speed of nearly 34 knots, and a mean speed for four hours of 33.1 knots, the vessel being run at full load. This performance makes her, for the time being, the fastest vessel afloat, and following on the failure of the sister ships of the *S 138* class to attain their designed speed, has created considerable interest in German naval circles where the turbine has hitherto been greatly disparaged. The main turbines consist of one high-pressure and two low-pressure cylinders of the usual type, arranged on three shafts, and revolve at about 850 revolutions per minute. Running astern, a speed of over 16 knots was obtained. The coal consumption averaged about 12 tons per hour at full speed.

No vessels corresponding to the large British destroyers of the *Cossack* type are as yet under construction, though it is possible that they may figure in the next Naval Estimates. For such speeds and displacements—33 knots and 780 tons—it is probable that oil fuel will be found essential, and this has not yet been applied to German torpedo craft.—"The Engineer."

The German Battleship Pommern has made a maximum of 20,348 horsepower, and 19.26 knots—a record for any German battleship so far, and well on the way to equal the French battleship *Liberté*. The coal consumption on the German ship was very high, while the *Liberté* accomplished almost a record in economy.

The speed of the Stettin is now stated to be 25.8. She is fitted with turbines of the Parsons type, and has proved about a knot better than her sisters, so far.

The German Navy League does not appear to be so satisfied with its Admiralty as is the British one. It is just conducting a strong agitation in favor of ships of 22,000 tons and 12-inch guns instead of 11-inch. Recent report has had it that the *Ersatz Sachsen* class are to carry two 12-inch per turret, instead of the original three 11-inch. The Navy League agitation suggests that the matter is still undetermined—a thing that should be useful powder to those of our contemporaries which expend so much energy in proving that the new German battleships do not yet exist.

Orders have recently been placed by the German naval authorities that indicate not only a remarkable change in their extremely antagonistic views of recent years on the subject of marine turbines, but also a genuine desire for definite information as to the relative values of different types of turbine. Cruiser "F," which is the first real continental reply to the British *Inflexible* class, has just been definitely ordered from Messrs. Blohm & Voss, of Hamburg, who have recently completed a splendid new machine shop for the construction of marine turbines. This vessel is to be completed within two years from now, and will prove a valuable addition to the German Navy. Her designed displacement is practically 20,000 tons, and her speed of 24½ knots will require about 44,000 indicated horsepower, which will be provided by turbines of the Parsons type. Exceptional care is being taken to keep the details of construction confidential; the main dimensions, however, appear to be as follows: Length on water line 560 feet, beam 85 feet, draught 27 feet.

Two small cruisers of 4,300 tons have also been ordered from the Vulkan yard at Stettin and from Messrs. Schichau, at Dantzig. The former will be fitted with a modified form of Curtis turbine supplied by the Allgemeine Elektrizitäts Gesellschaft, and the latter with turbines of the type introduced by Melms and Pfenniger. The latter turbine is a modified

Parsons, embracing many features of the Westinghouse combination system. The Krupp Germania works are also reported to have the order for a similar vessel which is to be fitted with Zoelly turbines. The speed in the case of these small cruisers is to be 25.5 knots.

GREECE.

Lonhi.—On July the 10th, in the presence of a large gathering, including official representatives of the Greek community in Great Britain, a torpedo-boat destroyer built for the Greek Government, was launched from the yards of Messrs. Yarrow & Co. (Limited) at Poplar, and received the name of *Lonhi.* The Greek Minister, who named the boat on behalf of the King of the Hellenes, was accompanied by Mme. and Mlle. Metaxas, and the staff of the Legation. After the religious ceremony by the Rev. Great Archimandrate Pagonis, the Greek Minister made a speech suitable to the occasion, and the launching took place amid the cheers of the company. The dimensions of the boat are: Length, 220 feet; breadth, 20 feet 6 inches; and depth, 12 feet 4 inches; I.H.P., 6,000; contract speed, 31 knots. The *Lonhi* is the third vessel of this class built by Messrs. Yarrow & Co. (Limited) for the Greek Government.

ITALY.

Much mystery still surrounds the Italian cruisers *Pisa* and *Amalfi.* These two ships of the *St. Georgio* class have been pushed on much faster than usually happens in Italy. The *Pisa* took the water at Leghorn recently, and the *Amalfi* is about to be launched at Genoa. The armament is four 10-inch, eight 8-inch; the horsepower 20,000. Belleville boilers have been ordered for them, not Blechyndens as originally reported. The *St. Georgio,* the first of the type, is apparently the only one with Blechynden boilers.

JAPAN.

Explosion on Japanese Battleship.—A terrible explosion occurred on the Japanese battleship *Kashima* on September 16th, while that vessel was at target practice off Kabutoshima in the

inland sea. The explosion, it is said, occurred inside a turret in which was a 10-inch starboard gun. The casualties first reported as the result of the explosion were: Killed, five officers and twenty-two men; severely wounded, two officers and six men; slightly wounded, two officers and six men. According to a later cable despatch from Mr. Dodge, Chargé d'Affaires of the American Embassy at Tokio, the explosion was more serious than at first reported. Mr. Dodge reports the loss of thirty-one lives instead of twenty-seven, and the injury of eleven men, and says that the disaster was "an explosion of shells." Naval Constructor Shinowara, Midshipmen Hayakiiwa and Nishimura were among those instantly killed. Lieutenants Arita, Fukuhara and Shigyo, and Midshipman Kitayama were said to be among those fatally wounded. After removal to the Kure Naval Hospital, Lieutenant Arita and eleven of the petty officers and men died. The Navy Department reports state that neither the gun nor the ship is seriously damaged. The *Kashima* is one of Japan's newest battleships, of 16,000 tons displacement, and was recently built in England. Among the officers and men killed were a number who had served with distinction in the Russian war. President Roosevelt at once directed the State Department to prepare a letter to Viscount Aoki, the Japanese Ambassador, expressing his sorrow at the news of the disaster on board the *Kashima*. Secretary Metcalf sent this despatch to the naval attaché at Tokio: "Convey to Minister of Marine the expression of the United States Navy's deep sympathy for the loss of life in the *Kashima* accident." The following official statement was submitted by Mr. Miyaoka to the Navy Department: "On September 16th, during target practice on board H.M.S. the *Kashima*, an explosion of charge occurred in the turret of the starboard aft 10-inch gun. The cause of the explosion is not clearly ascertainable owing to the fact that all officers and men on the spot have either been killed or severely wounded, but according to the statement of a man who left the turret just a moment before the explosion took place, it appears that when the accident occurred the gun in question had already been

twice fired. A new projectile was rammed and the charges were placed in position for the third fire, but the breech block had not been closed, when the said charge, catching fire from the backfire, set on fire the charge for the fourth projectile which was at the back of the gun. The projectile remained in the loading position. There was but little damage done either to the ship, the gun or the carriage. The number killed is: Officers and warrant officers, seven; petty officers and men, twenty-seven. The number of wounded, eight."—"A. & N. Journal."

SPAIN.

The Spanish Navy.—The reorganization of the Spanish Navy, which, according to official lists, only comprises one battleship, two armored cruisers, and six protected cruisers, besides five torpedo-boat destroyers, some gunboats and torpedo boats, will not only consist in the building of a number of up-to-date warships, but will also include the extension and reconstruction of the naval ports, and the construction of new naval fortifications. When the naval ports have been duly extended and enlarged, it is proposed to build the large ships in Ferrol, the smaller in Carthagena, and Cadiz is to have a large arsenal, more especially as far as artillery is concerned. Each of the three ports in question is to have its own floating dock. The ports of Vigo, Mahon, Ceuta, and Melilla will be fortified, but it is not proposed to make them arsenals. As far as the new fleet itself is concerned, the plan is understood to comprise six battleships, of 16,000 tons each, ten fast smaller cruisers, and twelve torpedo-boat destroyers. The ships first taken in hand will be three battleships and three torpedo-boat destroyers, and the armament will, it is understood, be specially adapted for coast defence. The reorganization of the fleet, and the works connected with it, will necessitate a State loan of 430,000,000 pesetas, which it is proposed to raise by an internal loan. The naval budget for next year will, according to the present plan, be increased from 35,000,000 pesetas to

50,000,000 pesetas. A considerable amount of trouble is anticipated in connection with the manning of the new fleet, and the English system of training is likely to be adopted.

SWEDEN.

The new Swedish battleships will be of 7,500 tons. Several are contemplated. Originally they were reported to be designed for four 12-inch guns, but now this has been changed. The present design is four 11-inch, four 7.6, eleven 4-inch, four one-pounders, and two 18-inch torpedo tubes. The belt will be 8 inches thick. The horsepower will be 17,500, and the speed 21 knots. Yarrow boilers will be fitted. Normal coal is only 450 tons, and the maximum capacity but 800 tons. The 7.6 guns seem rather superfluous, as they are neither one thing nor the other. An armament of six heavy guns would, we think, have been preferable.

The increased size of destroyers appears to be universal. Sweden is at present in the forefront in this direction, excluding the British *Swift*. The new Swedish destroyers are to be of 835 tons, 30 knot speed, and an armament of six 4-inch guns. The coal supply is very large.

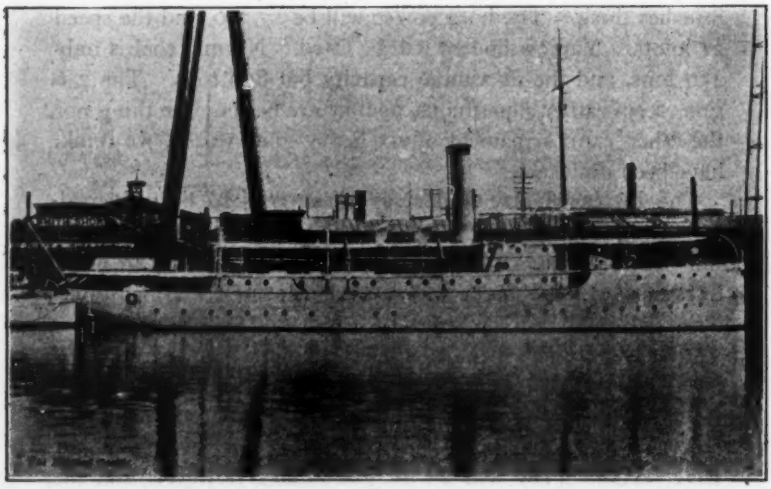
This quite eclipses the new German destroyers of the "V" type, of which twenty-four are ordered or being built—twelve for the 1906-07 program, and twelve for the current year. They are 520 tons only—that is to say, about the size of our River class. They will all be 30-knotters, however, and the armament one 24-pounder, three 6-pounders, and three tubes.

The armament of the Swedish destroyers, *Mode* type, is being altered from one 12-pounder, and five 6-pounders, to four 12-pounders.

UNITED STATES.

Revenue Cutter Pamlico.—The new revenue cutter *Pamlico,* built by the Pusey & Jones Co., Wilmington, Del., for the Revenue Cutter Service, is for the North Carolina sounds. She was recently placed in commission at Baltimore, with First Lieutenant H. G. Fisher in command, and Second Lieutenant J. L. Maher as executive officer.

The *Pamlico* is a shoal-water, single-mast, twin-screw steamer, and cost in the neighborhood of $175,000. The principal dimensions of the steel hull are as follows: Length over all, 158 feet; length between perpendiculars, 148 feet; depth from base line at side, amidships, 10 feet; spring of beams in 30 feet, 7½ inches; breadth of beam, molded, 30 feet; draught of vessel with 25 tons of coal and 2,400 gallons of fresh water, stores and provisions, ready for sea, 5 feet; displacement at 5-foot draught, 408 tons.



NEW REVENUE CUTTER "PAMLICO," BUILT BY THE PUSEY & JONES CO., WILMINGTON.

The hull is divided into six compartments by five watertight bulkheads. On the top gallant forecastle is located the steam windlass engine, anchor davits, and a 6-pounder rapid-fire gun. From this deck a hatchway and ladder afford access to the main deck. The top gallant forecastle deck extends aft to the top of the deck house, thus forming a continuous deck, affording protection to the main deck forward.

There are two triple-expansion propelling engines of the vertical, inverted cylinder, direct-acting, type, each having one

high-pressure cylinder 8 inches, one medium-pressure cylinder, 13¼ inches, and one low-pressure cylinder, 22 inches in diameter, the stroke of all pistons being 18 inches. There will be one Sirocco fan blower driven by steam turbine, located in the boiler compartment. It is designed to run with a steam pressure of 100 to 120 pounds, and will be connected with ducts to the ashpits, and to the ventilating system of the vessel. Steam is supplied from a boiler of the marine water-tube type, 9 feet 3 inches in length, 11 feet 9 inches in width and 11 feet in height, working at a pressure of 200 pounds to the square inch. The boiler for the donkey engine is of the Scotch type, with a working pressure of 200 pounds to the square inch.

The vessel has only one mast, which is 64 feet in length over all, fitted with yard arms and a boom, for wireless telegraphy.—"The Nautical Gazette."

MERCHANT SHIPS.

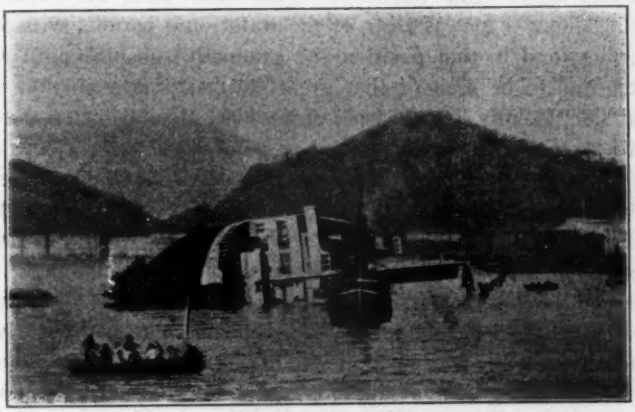
The Disaster at Riva Trigoso, Italy.—A feeling akin to consternation has been spread through Italian shipping circles owing to the mishap in launching the biggest passenger steamer yet constructed in Italy. The building yard is situated at Riva Trigoso, a small place about halfway between Genoa and Spezia.

It was started in 1889, and has gradually increased until now it has an area of 40,000 square meters, and three building slips of masonry, capable of taking vessels of 150 meters (492 feet) in length. These slips are served by seven radial electric cranes capable of lifting a weight of 2 tons to a height of 25 meters, with a radius of 20 meters, which were described in one of the papers read at the last session of the Institution of Naval Architects.

The size of the vessels constructed has gone on increasing, until the last to be laid down were the biggest yet constructed in Italy. These two vessels, to be known as the *Principessa*

Jolanda and *Principessa Mafalda*, were for the South American service of the Societa Lloyd Italiano. They are 149 meters long (489 feet), with a beam of 17 meters (56 feet), and were to have a displacement of 12,000 tons. They have twin-screw quadruple-expansion engines of 10,000 horsepower, which were to give the vessel a speed of 18½ knots. The first of these two vessels—the *Principessa Jolanda*—was finished complete, with engines and boilers, and was to be launched on September 22d, whilst the second vessel is well advanced, all her framing being completed.

The day fixed for the launch was a typical Italian day—



bright sunshine, a cloudless sky, and delightful temperature. The trains from both directions of the Riviera took hundreds of eager spectators to the place, and many steamers, large and small, took other visitors down from Genoa, and the sea being quite smooth, these were largely patronized.

Shortly after midday the naming took place in due form, and the bottle of sparking asti, decorated with flowers and the Italian colors, was broken in the traditional manner. Twenty minutes afterwards the signal was given that all was clear, and the vessel gradually began to move, quickly increasing her rate of speed, and amidst the cheering of thousands, the hoarse

tones of the whistles, and the shrieking of syrens, the launch was quickly and triumphantly completed.

But no sooner was the vessel fairly afloat than she was seen to keel over in an alarming manner; the cheering ceased in an instant, and a dead silence followed; the effect of this and the huge mass of the vessel slowly going over was so horrifying to the spectators that they started to flee from the spot.

The tugs had at once got hold of the vessel, and she was pulled round parallel to the shore. The inclination was to port, and by this time the water had reached the port-holes of the main deck, some of which had been left open, and the cabins quickly filled. She was soon on her beam ends, her funnels being about 2 meters clear of the water, and parallel with its surface, and in that position she gradually subsided until all that was to be seen of that great steamer was a portion of her side, looking like the back of a whale, about a meter and a half above the surface at its highest point.

The effect on the spectators was intense. The shipyard is so situated in the corner of the bay that the vessel seemed to dominate everything. That she should have entirely disappeared in such a fashion was horrifying to the onlookers; it seemed as if some terrible nightmare had got hold of them, and they could hardly speak. The officials who had been responsible for the construction stood there as if turned to stone, and remained gazing at the spot as if they could hardly believe their eyes, whilst the workmen who had built her, and who, with their families, had almost cheered themselves hoarse a few minutes ago, now were weeping and hugging one another in a state bordering on delirium.

Bad as the disaster was, it was not rendered still worse by loss of life, for the disappearance was so gradual that all on board were got off before the hull went under.—"Engineering."

A Motor-Driven Liner.—There is now under way in England an experiment which, if successful, will mark a new step in marine propulsion and achieve results by which the *Lusitania's* speed record will be put in the shade. The keynote of

the idea is the application of electricity to turbines, and a well-known firm of engineers is equipping a vessel with an apparatus designed to make the test both practical and complete.

It must be remembered that the steam turbine is most efficient when running at high speed, while a ship's propeller, on the other hand, will not work efficiently at the highest speed. If the speed be increased beyond a certain point, far below the most efficient speed of the turbine, the blades of the propeller simply churn the water instead of driving the ship. It is impossible to gear down from a turbine to a propeller shaft, for the horsepower of marine turbines is too great for any practicable form of gearing. Consequently the turbine has to be run slowly, and an inevitable loss of efficiency in this direction is put up with.

The speed of the *Lusitania's* turbines is only 180 revolutions per minute, and to adapt them to these conditions they are large in diameter and have blades of great sectional area. This means that there must be sparse clearances, and these in turn mean that steam entering the turbines at high pressure finds its way toward the condenser without giving out the whole of its heat and energy. If brought to perfection electrical transmission will form a link between the swift turbine and the slow propeller.

The plan upon which the firm of engineers which is now preparing to make the practical test spoken of is not that the turbine should be coupled directly to the propeller shaft, as is now done, but should drive high-speed electrical generators and supply current to electrical motors for driving the propellers. A qualified engineer further explains the idea as follows:

"For fast passenger boats the arrangement will resemble that of a modern electricity-supply station with the many units of the plant feeding the common system of mains. All the turbo-generators will supply the common bus-bars on the switch-board, so that it will be possible to feed any motor from any generator. The system of supply adopted will be either continuous or alternating, preferably the latter, as high pressures

can be used and commutation troubles avoided. Some alteration in the disposal of the machinery would be necessary, but on the whole there would be a gain of space. But more important than any consideration of space, the electrical system possesses the advantage that the motors can be reversed almost immediately."

A future *Lusitania* may be driven by turbo-generators of 100,000 H.P. at a speed of 30 knots. Such a vessel would have six turbo-generators of 20,000 H.P. each, one of which would be in reserve. Each of her four propellers and the shafts would be provided with six motors of 5,000 H.P., five of which would do the work while the other would be a standby, running light but ready on the pressure of a button on the bridge to take up its share of duty.

For the bridge electrical transmission will mean a revolution, the navigating officer will no longer have to signal his orders for the maneuvering of the ship to the engine room. He will have beside him a keyboard of push buttons by which he himself will control every movement of the ship instead of ordering the engineers. To go astern, for example, he will push a button which will reverse the motors, and so with every variation of speed and direction.

Such an accident as occurred to the *Deutschland* in Dover Harbor would be impossible. The eye that sees the danger and the hand that prevents disaster will be controlled by one brain, and the navigating officer on the bridge, conscious of imminent peril, will not have to transmit mechanically his orders to the unseen engine room below, where their immediate performance, on which the vessel's safety depends, may be hindered by slow comprehension or an accident of some other nature.

In the electrically-propelled ship the eye that sees and the brain that understands will alike control the propeller and the helm.—"The Marine Review."

MANUFACTURERS' NOTES.

WATERPROOF AND STEAMPROOF LEATHER BELTING.

Waterproof and steamproof leather belting was exhibited, for the first time in history, at the Jamestown Exposition. Charles A. Schieren & Co. have received a gold medal, the highest award, from the exposition for their Duxbak Leather Belting on the distinct claim that it is a waterproof and steamproof belting. Charles A. Schieren & Co. have been perfecting their waterproof belting for several years but never until about a year or two ago was it advertised under a trademark name and with the distinct claim that it was waterproof and steamproof. They claim that a distinct and important step forward has been taken in the making of leather belting in their Duxbak process.

The exhibit of the Schieren Company at Jamestown has attracted widespread attention throughout the months of the Exposition. It is a beautifully-painted glass picture representing a lake, out of which flows a real waterfall upon a roll of Schieren's "Duxbak" belting. The exhibit is designed on the lines of an interior view of a camera. From the front of the booth black cloth is draped artistically on the two sides and top to the rear of the booth, throwing the brilliantly-lighted picture out into bold relief. The size and magnificence of this display can best be realized when it is known that the beautiful big roll of "Duxbak" belting is eight inches in width and forty-two inches in height. The rear of the plate of glass and the frame are lit up with 148 electric lights. Appearing and disappearing signs occupy either side of the display. On one side the "Duxbak" trademark appears, followed on the other side by the words, "Chas. A. Schieren & Co., New York."

The water falls from a fissure in the glass so naturally that it appears a natural waterfall from the lake itself. It

drenches the entire surface of the belt coil, and the pond formed by the overflow covers the bottom part of the coil to the depth of eight inches. Notwithstanding the fact that the belt is soaked all day long in this depth of water in the tank, it is perfectly dry and flexible when the water is turned off at night and the tank drained. In fact, the belting is so firm and solid and has so little affinity for water that it hardly appears at all wet the instant the falls are turned off. In order to further demonstrate the absolutely waterproof character of the "Duxbak" belting the end of the roll is uncoiled and is erected under the falls so that the sides of the roll are thoroughly drenched. The water hits the belt just at the end of the lap, and plays continuously all day long right in the cement—a test to which no belt has previously been submitted. This particular coil of belt has already remained under the water 5½ months and it is the intention of Chas. A. Schieren & Co. to leave it there during the remainder of the Exposition, having confidence that it will be as good as new at the end of that time. This is certainly convincing testimony as to the waterproof character of "Duxbak" belting. Visitors are greatly impressed by the substantial and honest character of this test, and everyone declares that in the line of belting the "Duxbak" has proved the best thing they have seen so far.

This is, in fact, the severest test to which a belt can be subjected. When it merely revolves in water the tendency is to throw the water off the surface of the belt, but with the waterfall under high pressure, as in Schieren's demonstration, the water strives to enter every fiber of the leather, but without avail, since Schieren's waterproof dressing sheds it like a duck's back, and the waterproof cement makes the laps impervious as well.

The balance of this electric illumination is also very attractive. The water in the tank is bordered by a semicircle of palms and ferns. On both sides of the booth, extending from the tank to the front of the booth, are arranged symmetrically fourteen rolls of the Schieren belting.

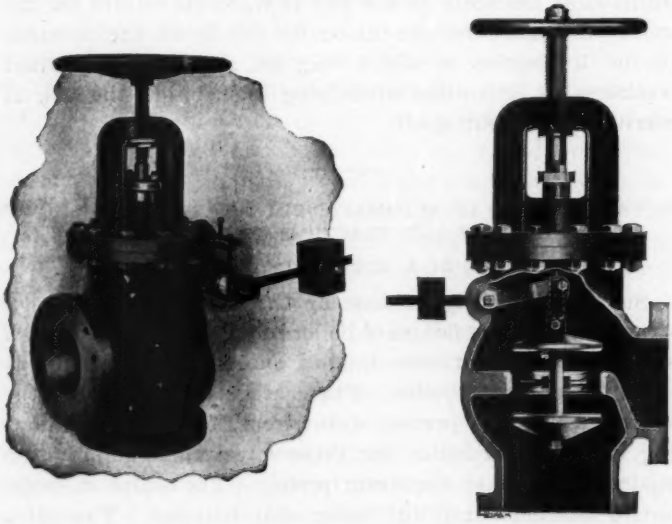
The exhibit is indeed unique and clever in a number of ways, and might be taken as a good object lesson by those advertising men who say Expositions cannot be made to pay. It has movement and life, the first of all requisites to draw a crowd. It has artistic merit; it has color; it makes prominent the trademark of "Duxbak" belting, all most important considerations. Best of all, however, it convinces the crowd and makes a lasting impression by its clear argument when the crowd has been attracted. The vast majority of such exhibits most assuredly do not pay in traceable returns for the money invested—but the reason for this is not the poorness of the Exposition at which they are shown, or any other weakness in Exposition advertising as such, but the lack of merit of the exhibit itself.

ADVANTAGES OF AN AUTOMATIC CUT-OFF VALVE AS SHOWN BY TESTS AND PRACTICAL EXPERIENCE.

By A. Eugene Michel.

Such great damage is done by the occasional bursting of steam mains or the failure of boiler tubes, where there is high steam pressure, that some kind of excess-flow safety valve is essential for every boiler. There are many devices on the market which will prevent steam from the mains from flowing back into the boiler, but these valves make no provision against accident to the steam piping. The escape of steam would continue until the boiler were emptied. The valve which we are about to describe not only acts as a "non-return" valve, but it also shuts off the boilers with equal surety and protection if the main should be broken, as from water hammer or from the breaking of an elbow. As a cut-off valve it acts automatically, but by the turning of a hand wheel it may further be used as an ordinary stop valve. As steam is raised in the boiler the valve opens it to the main when the boiler pressure is equal to the pressure in the line, thus avoiding all accidents from carelessly opening valves, while there is considerable difference of pressure.

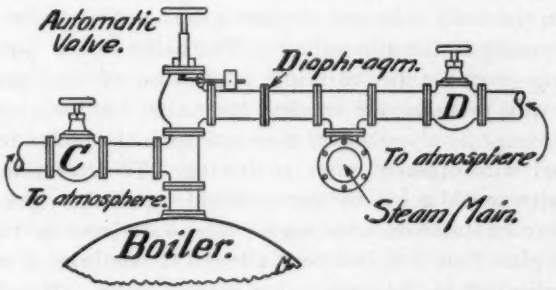
This valve works instantly either way and does not depend upon difference in pressure for its action, but upon the actual flow of steam through the valve. If a tube in one of the boilers should give way it shuts down that boiler only, and allows all the other boilers in the battery to go on supplying steam as usual. With an ordinary stop valve the fireman is frequently scalded in trying to shut off the main or is driven from the fireroom without being able to do anything to save the entire plant from shutdown. This cut-off valve will, un-



less intentionally opened, stay closed until the pressure is raised again. It is, therefore, perfectly safe for a man to go inside to repair damages as soon as the injured boiler has cooled off sufficiently.

If, on the other hand, a steam header bursts, or a joint breaks or a cylinder head blows off the engine, the cut-off valves on all the boilers close immediately before the room has been filled with steam, and repairs can proceed at once. The operation of this valve may easily be seen from the accompanying cross-sectional view. It is installed so that the

lower valve disc is toward the boiler and the hand wheel on top. Normally, when the boiler is not working the upper valve rests upon the seat and prevents steam from the main from entering the boiler. When the steam pressure in the boiler is raised to slightly exceed that in the main, the valve lifts and steam flows from the boiler into the main. The valve is very nearly counterbalanced by a weight on an external arm. The leverage of this weight may be adjusted to different rates of steam flow for boiler output. The valve is operated, not by pressure, but by the actual flow of steam through it. The normal flow of steam into the main raises the discs to mid position, as shown. The valve remains in that position as long as steam is being drawn from the boiler



at the normal rate, but in case of a break on line side of the valve the excessive rush of steam would carry the lower valve up against the seat, shutting off the boiler. Of course, when the flow reverses the upper valve drops instantly to its seat and shuts off the steam. The rate of flow at which the boiler would be shut off is determined by the weight above mentioned, and by the distance between the two valve faces. This is adjusted to correspond to the greatest overload at which it is desired to operate the boiler, say a rate over twice the normal rated capacity, or when the water begins to raise in the gauge glass.

A fork-and-link arrangement connects the valve discs and balancing lever positively, so that the position and condition

of the valve may be determined by a glance at the balancing lever.

The balancing lever is provided with light springs which prevent the chattering or closing of the valve under ordinary conditions. The springs are not strong enough to prevent closing in case of accident, but will prevent closing in case of a momentary rush of steam. These springs can be adjusted to meet different quantities of flow of steam.

The following tests made on this valve in the power house of the Anaconda Mines, Anaconda, Mont., illustrate how it acts in case of accident. The accompanying illustration shows the arrangement for the test. The automatic valve was placed between the boiler and the main. On the boiler side was a quick-opening valve "C" communicating with the atmosphere, and on the main side was another quick-opening valve "D," also opening to the atmosphere. The boiler was a 300-H.P. Stirling water tube, carrying a pressure of 150 pounds. When this pressure was reached the valve "D" was opened wide enough to allow steam to escape from the boiler to correspond with different rates of driving. To determine how the valve would act if the main should burst, the valve "D" was thrown suddenly wide open. The water rose so rapidly in the glass that if it had been allowed to continue, it might have resulted in the destruction of the boiler. Before the water had gone up two inches, however, the automatic valve closed tight. A test was then made to see if the automatic valve would close if the break opened slowly. A steel diaphragm was placed in the pipe as shown; in this diaphragm was an opening $1\frac{3}{8}$ inches in diameter, and when the valve "D" was opened suddenly the automatic valve closed promptly. The diaphragm was then removed and another substituted with an orifice $1\frac{1}{2}$ inches diameter. When the valve "D" was opened again the automatic valve did not close, showing that its point of closing was between these rates of delivery. In this case it was set to close at a flow corresponding to 600 boiler H.P.

After making these tests the boiler was again placed in

service and a test was made to see what would happen if a boiler tube should break or some part of the boiler give way. This was done by suddenly opening valve "C," upon which the automatic valve closed promptly, shutting off the main. Fifty-four tests like these were made and the valve acted perfectly each time. The tests were so satisfactory that sixteen of these valves were installed in the plant.

The manufacturers frequently receive reports of accidents where the valve more than pays for itself in one emergency. One of these valves had hardly been installed in a plant recently when a blind end blew off the steam header. The engineer had forgotten to put in all the bolts, but the automatic valve acted before further damage could be done.

Bursting steam pipes frequently scald their victims to death where there is no automatic means of shutting off the flow, but the danger in refrigerating plants is even greater. Just what ammonia will do if the flow is not stopped by a cut-off valve may be judged from accidents which have happened at Armour's Refrigerating Plant in Chicago. A cylinder head blew off an ice machine last January while twenty men were in the room. Three were killed, sixteen overcome by the fumes before they could get out, and only one escaped. In May a two-inch ammonia pipe burst, and the fumes killed six men, 200 head of cattle and a thousand sheep. The ammonia fumes were so strong that rescuers could not enter for four hours. A cut-off valve in the line would have checked the fumes in both instances before enough would have escaped to be serious.

The Lagonda automatic cut-off valve is made by the Lagonda Manufacturing Company, Springfield, Ohio, who will be pleased to send further descriptive literature upon request.

THE NEW PLANT OF THE LAGONDA MANUFACTURING COMPANY.

Twenty years ago, when water-tube boilers first began to be introduced, a man named Weinland invented a machine for cleaning scale from out the tubes. It was of the type now

known as the "turbine." When these cleaners were put on the market the business was carried on in a 10-foot by 12-foot room, the cleaners being made outside. The venture proved successful, and the cleaners came into favor so rapidly that machinery for their manufacture was purchased and installed in a room 24 feet by 24 feet. The company then began to take contracts to clean boilers. In that way the merits of the cleaners were demonstrated to possible customers, and those who did not care to bother with the cleaning themselves were assured of having the work done quickly by competent men. Again the quarters became too small. This time the plant was moved into a 28-foot by 70-foot two-story iron-frame building (built by the present company), and every effort was made to build the most and the best cleaners used in this country.

As water-tube boiler plants became larger and larger the scale problem assumed more importance, and the time that boilers had to be laid off for cleaning represented a heavy loss of revenue, so the Lagonda Manufacturing Company, the concern referred to above, sought to devise some machine that would permit the application of greater power to the scale-cutting device than was possible with the turbine cleaner. The result was their Weinland Mechanical Boiler-Tube Cleaner, which can be propelled through the tube by any power, from a 4-H.P. water motor to a 25-H.P. steam engine or electric motor.

Where scale is exceptionally hard or thick, or where there are so many boilers that cleaning becomes a heavy item of expense, or where the water for driving turbine cleaners would cost considerable, the power-driven mechanical cleaner is by far the most economical device that can be used. It is also much cheaper to maintain, as its parts are almost indestructible and are cheaply replaced. The Weinland Mechanical Cleaner is used almost exclusively by the Lagonda Company in its boiler-cleaning contract work.

The Lagonda Manufacturing Company has also added other specialties and, with the increasing demand for the Weinland

cleaners, has found greater manufacturing facilities necessary. A new plant has therefore just been built, consisting of a two-story 50-foot by 200-foot building, located on two acres of ground, which will be available for further growth.

The new building, although not first in size among modern factories, is a model in the character of its construction and equipment, and it is undoubtedly the largest plant in the world building boiler-tube cleaners. The walls and floor are of artificial stone, and both floors are well heated, lighted and ventilated. It has modern wash rooms, lockers and toilets. Every machine tool is driven by an individual motor receiving power from a gas-engine-driven generating plant. The absence of line shafts and belts and the large windows give plenty of light and make the shop an ideal place for workmen to do their best.

All the lathes and automatic machines are made with special attachments for turning out the peculiar bearings, water wheels, arms and other small parts accurately and in great quantities. The equipment includes furnaces for heating oil and lead baths for tempering an elaborate testing bay, for trying out each finished cleaner before allowing it to leave the factory. This testing department contains tanks, pumps and gauges of proper size to subject every cleaner to actual working conditions.

The cut-off valve department, is devoted to the manufacture of a device placed in the steam connections of boilers for automatically stopping the passage of steam in either direction in case a steam main bursts or a boiler tube gives way. The sales of this valve seem to indicate that engineers are quick to appreciate the protection which it affords. Many "non-return" valves have previously been offered to steam users, but this is the first practical one which works equally well in both directions and which contains no pistons or other sliding parts which could stick or bind.

FREE ENGINEERING LIBRARY TO OPEN EVENINGS.

On and after Wednesday, November 6, 1907, the reference libraries of the American Institute of Electrical Engineers, The American Society of Mechanical Engineers and The American Institute of Mining Engineers, 29 West 39th Street, New York, will be open evenings until nine o'clock on all week days except public holidays.

These Libraries, constituting practically one library of engineering, situated near the New York Library, in the new headquarters of the Engineering Societies, are available to members of the above Societies, engineers, and the public generally, subject to proper regulations. Strangers are requested to bring letters of introduction from members or to secure cards from the Secretaries of the respective Societies.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The fifty-fourth annual meeting of the American Society of Mechanical Engineers will be held in the Engineering Societies' Building at 29 West 39th Street, New York, December 3–6, 1907.

Symposiums on foundry practice, giving the experiences of prominent men in that work, have been arranged. The specific heat of superheated steam will be taken up, a very important and exhaustive work by a Professor of Engineering at Cornell, will be presented. The utilization of low-grade fuels in gas producers, combustion control in gas engines, tests of producer-gas engines, etc., will be given a session. Other live topics, such as industrial education, power transmission by friction driving, cylinder port velocities, etc., will be discussed.

All of these subjects have been treated by prominent engineers of Europe and America, professors of our universities, and men eminent in the particular work of which they write.

The Committee have on hand an interesting excursion for Wednesday afternoon and an address in the evening which will be especially enjoyable.

BOOKS RECEIVED.

ELECTRICAL INSTALLATIONS OF THE UNITED STATES NAVY. A manual of the latest approved material, including its use, inspection, care and management, and method of installation on board ship. By COMMANDER BURNS T. WALLING, U. S. Navy, and JULIUS MARTIN, E. E., Master Electrician of the Equipment Department, Navy Yard, New York. Published by the U. S. NAVAL INSTITUTE, Naval Academy, Annapolis, Maryland. Price $6.00, post paid.

This very excellent work, consisting of 648 pages, gives a most comprehensive description of the electrical apparatus in the Navy, together with full information as to the inspection, care and handling.

Its value to those who have charge of the electrical machinery on board our naval vessels cannot be overestimated. The student will find it equally valuable. The book is divided into fifteen chapters, under the following heads:

Chapter.

I. Incandescent Lamps.
II. Arc-Lamps and Search-Lights.
III. Standard Wire.
IV. Wiring Appliances.
V. Generating Sets.
VI. Generating Sets (continued).
VII. Motors.
VIII. Motors (continued).
IX. Miscellaneous Motor Applications.
X. The Inspection of Generating Sets and Motors.
XI. Auxiliary Apparatus and Instruments used with Generating Sets and Motors or for Tests.
XII. General Notes on Generating Sets and Motors.
XIII. Electric Fixtures and Lanterns.
XIV. Interior and Exterior Communication.
XV. Notes on Installation.

MODERN MARINE ENGINES WITH THEIR AUXILIARIES AND APPURTENANCES, LAUNCH MOTORS AND STEAM TURBINES. For educational and practical purposes. By ROSENTHAL, MUELLER and BAYER, Instructors of Engineering at the School of Navigation, Hamburg, Germany. K. M. MECKLENBURG, Berlin, Publishers.

The value of this interesting work is greatly enhanced by a volume of 53 plates, lithographed in colors, containing over 1,200 illustrations of machines. Its size, 8 inches by 11 inches, is very convenient and handy for reference since there are no folders.

The work throughout contains only practical material. The drawings are neatly executed, without unnecessary complications by figures, and the tinting of the sections in colors denoting the materials, adds not only to the appearance but also to the clearness of reading. This method is certainly much to be preferred to the usual cheap and clumsy methods of cross hatching in one color, and is well worth the additional expense.

Originally intended only as material for the course of instruction and with the intent to assist the student in the preparation for his examinations, this book has also developed in a large measure as an *aide de memoire* and book of reference for practical engineers, besides giving many constructions that may be new to some.

The plates contain illustrations of cylindrical, water-tube, locomotive, donkey and launch boilers, with all their details. There are drawings of furnaces, bracing, stacks, of all usual fittings, as gauges, reducing valves, hydrokineters, circulators; various methods of burning fuel oil and coal are illustrated. Ash injectors, blowing engines, the Howden and the Ellis & Eaves system of forced draft are shown and feed-water filters and heaters are not forgotten.

The construction of the engines is fully illustrated by full details of all parts, and also by several examples of engines in actual use, and from the designs of notable builders. They embrace all types, and vary from a small launch engine to the

engines of the *Deutschland*. Our much-admired framing of the engines of our destroyers is given a prominent place. There are full diagrams of crank efforts, inertia, indicator diagrams, theoretical, actual and combined.

There are full drawings of the necessary auxiliaries, many types of pumps, ejectors, turning and steering engines, ash ejectors, etc.; in fact, all that is necessary for an engineer of a vessel to be familiar with.

Of no little interest are the drawings of various gas and oil engines for launches, as well as the principal systems of steam turbines.

The entire work bristles with interesting data and is a most valuable adjunct to any engineer's library.

The authors deserve great credit for the selection of the most valuable material on the subject of marine engineering.

SCHIFFSKESSEL. By WALTER MENTZ, professor in the Royal Technical School of Danzig. Munich and Berlin: R. OLDENBOURG, 1907.

This book of 300 pages, treating of the development, the design, the construction and the arrangement of boilers on board ship, is a valuable addition to the literature on the subject of marine boilers. The portion devoted to fire-tube boilers is very comprehensive, and the illustrations excellent. That treating of water-tube boilers gives the Schulz-Thornycroft in considerable detail, but only short descriptions, with illustrations, of other Continental and British boilers, while mention is made of only two boilers of American design, the Babcock & Wilcox and the Stirling, the latter of which, as stated by the author, is not in use afloat. Notwithstanding this omission, and the disappointing nature of the chapters devoted to water-tube boilers, the book possesses decided merit. The text is in German, and the measurements in metric units.

OBITUARY.

It is with much regret that the Council announces the death of one of its members since the last issue of the JOURNAL.

Rear Admiral WILLIAM A. WINDSOR, U. S. N. (Retired), died on August 30, 1907.

Admiral Windsor was born in Virginia and was appointed a Third Assistant Engineer in the Navy in 1862. He was transferred to the retired list September 16, 1902, on his own application, after forty years' service.

NOTES OF THE SOCIETY.

A regular meeting of the Society was held at the Navy Department, October 1st, at which nominations were made for officers of the Society for the coming year, as follows:

For President.

Commander W. M. PARKS, U. S. N.
Commander R. S. GRIFFIN, U. S. N.

For Secretary-Treasurer.

Commander THEO. C. FENTON, U. S. N., Retired.
Lieutenant J. B. GILMER, U. S. N.

For Members of Council.

(Three to be selected.)

Commander F. C. BIEG, U. S. N.
Commander B. C. BRYAN, U. S. N.
Commander H. P. NORTON, U. S. N.
Engineer-in-Chief CHAS. A. MCALLISTER, U. S. R. C. S.
Commander W. W. WHITE, U. S. N., Retired.
Lieutenant Commander C. W. DYSON, U. S. N.
Lieutenant J. B. GILMER, U. S. N.

Voting slips have been forwarded to all the members, to be returned at once. The votes will be counted at a meeting of the Society to be held at the Bureau of Steam Engineering, Navy Department, at 4.30 P. M., Thursday, December 26, 1907.